S. A. E. JOURNAL

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The purpose of meetings of the Society is largely to provide a forum for the presentation of straightforward and frank discussion. Discussion of this kind is encouraged. However, owing to the nature of the Society as an organization, it cannot be responsible for statements or opinions advanced in papers or in discussions at its meetings. The Constitution of the Society has long contained a provision to this effect.

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History Reviewed at Summer Meeting

Reunion of Society's Founders, Historical Exhibition, Program of Excellent Technical Sessions and Ideal Weather Combine to Make 25th Anniversary a Great Success

NEVER will the members of the Society who were so fortunate as to be able to attend it forget the 25th Anniversary Summer Meeting of 1930. Everything combined to make it a huge success from start to finish. Even the weather favored the meeting extraordinarily, for, unlike that of the last previous meeting at French Lick, when there were successive days and nights of torrential rain and hail and the hotel grounds and all the surrounding country were flooded, every day of this year's meeting was blessed with dry, sunny and unusually cool weather for the last week in May.

Seven of the founders of the Society, three of whom were Past-Presidents, were on hand, still hale and vigorous, to take part in the proceedings and to recount their experiences in the infant days of the automobile industry and in the organization of the S.A.E. at the 25th Anniversary Session on Wednesday night. These pioneers had a delightful reunion and throughout the week the anniversary spirit pervaded the meeting. All of the members were delighted to see those courageous and far-sighted men who so ably helped to launch the industry more than a quarter century ago looking and acting so well and vigorous. Though a number of them have passed three score years, their recollections of the experimental days of 30 years ago are very vivid and they are all physically fit, as was attested by the fact that E. T. Birdsall, the oldest of the lot and the father of the S.A.E., drove his Stutz car from his home at Hackensack, N. J., to the meeting in three days, accompanied only by Mrs. Birdsall.

The Historical Exhibit arranged for the meeting brought together the greatest assemblage of early motorcars, aeronautic engines, automotive parts, and early literature of the industry that has ever been collected and led to the general expression of sentiment that it should all be preserved in a museum. All of the historic cars, engines and parts are being preserved, of course, but in scattered places, as the Smithsonian Institution, the National Museum, Mr. Ford's museum at Dearborn, Mich., and the individual factories, by which they were generously loaned for the occasion; but the automotive industry would like to see all of the material permanently kept together in a general exhibition of American automobiliana.

The pageant that closed the meeting on Thursday afternoon, although composed mainly of Army motor-vehicles, also bore its historic significance, for the Class-B 3-ton trucks were the outgrowth of standardization and designing assistance rendered by the Society to the Government during the strenuous days when this Country became involved in the World War more than a dozen years ago.

Whatever regret was felt at the meeting was on account of numerous faithful members of the Society who have passed on and of other members usually seen at these annual gatherings who, for business or other reasons, were unable to be present at this celebration meeting in particular. Whereas the attendance at the Summer Meetings of recent years has ranged from 800 to nearly 1000, the registrations this year totaled only 690. This is attributed in large measure to the fact that the manufacturing plants are unusually busy so early in the season this year in preparation for the production season that promises to be very active through the coming summer and autumn.

All of the technical sessions, which were held only in the forenoons and evenings, leaving the afternoons free for outdoor recreation, were well attended, even the sessions on very specialized subjects drawing good-sized crowds and bringing out considerable discussion. It is perhaps significant and indicative of the probable future course of the Society that several of the founders and other speakers whose suggestions command respect expressed the belief that research will in the coming years become of major importance and that the Society should extend its relations and cooperation more broadly into all allied fields of activity.

Credit for the pronounced success of the meeting belongs to many who gave freely of their time and best thought to arrangements for and the conduct of the sessions and special events. Among these are the members of the Council, of the Meetings Committee headed by J. A. C. Warner, the Meetings Committees of the several Activities, the energetic President of the Society, the Chairmen of the various sessions, the authors of the numerous papers presented and the members who contributed much worthwhile discussion. To the list may well be added the thanks of the Society to those founders whose presence and addresses contributed so much to the enjoyment of the occasion and also to the many organizations that loaned exhibits and to the Quartermaster General's Department.

Summer Meeting Reports

Meetings Calendar

National Meetings of the Society

Chicago Aeronautic—Aug. 26 to 28

In Conjunction with National Air Races.

Production—Oct. 7 and 8

Book-Cadillac Hotel, Detroit.
Production Dinner—Oct. 8.

Transportation—Oct. 22 to 24

William Penn Hotel, Pittsburgh.
Transportation Dinner—Oct. 23

West Coast Transportation (Tentative)—November
San Francisco.

June Section Meetings

Baltimore Section—June 18

Baltimore Automobile Trade Association.
Subject—Does Self-Service Pay?

Chicago Section—June 10

Detroit Section—June 16

Book-Cadillac Hotel.

Northwest Section—June 21 and 22

Outing and Inspection of City Hydroelectric Plant at Rockport, Wash.

Oregon Section—June 28

Outing and Entertainment at Longview, Wash.

No Meetings Scheduled by

Canadian, Cleveland, Dayton, Indiana, Metropolitan, Milwaukee, New England, Pittsburgh, Southern California and St. Louis Sections.

Founders Recount the Early Days

Riker, Birdsall, Alden, Chatain and Wall Tell of Society's and Industry's Infancy—Vincent Bendix Heads Nominees for 1931

THROUGHOUT the 1930 Summer Meeting the 25th anniversary sentiment pervaded the sessions and the informal gatherings. The last evening session, on Wednesday night, was designated on the program the 25th Anniversary Session and will go down in the history of the Society as notable for the reunion of the founders of the S.A.E. As President Warner pointed out, exactly enough members of the Governing Board of the Society in 1905 were present at the session for the transaction of business as required by the Constitution of the Society, which provides that 30 per cent of the Council constitutes a quorum, and just three members of the first Board were on hand. Seven of the men who partook in the work of founding the Society and sat in its earliest councils quarter of a century ago were called to the rostrum, where they sat behind the speakers' table through the session as honored guests and were called upon in turn to give short reminiscences of the pioneer days in the industry.

Other notable features that made this session of extraordinary interest were the announcement of the nominees for office in the Society for 1931, to be elected at the Annual Dinner next January; the reading of a telegram of greetings and congratulations from President Hoover; the reception of an address transmitted by telephone and loud speaker from Alvan Macauley, president of the National Automobile Chamber of Commerce, from his home in Detroit; the conferring upon Secretary Coker F. Clarkson of the only honorary membership so far awarded by the Society; the reciting by Will Herschell, the bard of Indiana, of a poem entitled, The Meaning of S.A.E., and a number of other "little things" written by himself; and, finally, the showing of a large number of lantern slides of notable Summer Meetings of the past, of early American endurance runs, and of forerunners of today's automobiles extending back 150 years or more.

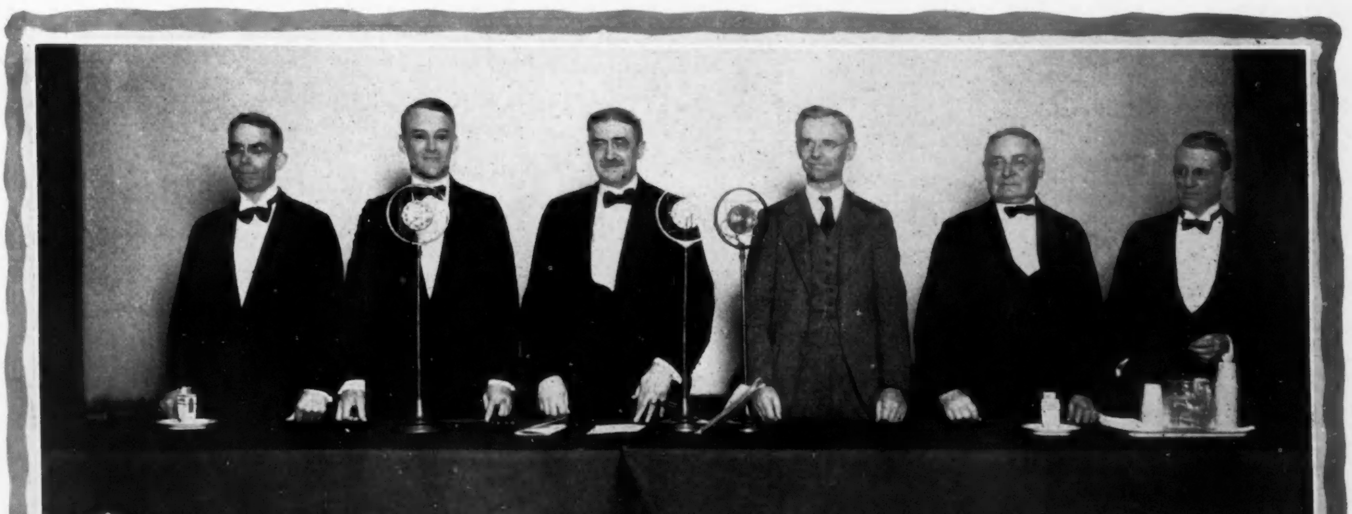
Looking Back to the Past

"This is an exceptional meeting," pointed out President Warner, in his opening address. "It is a meeting at which we look not only forward but also back to the beginning of the several branches of industry that the membership of the Society represents, to the days when confidence in the future of the automotive industry was the possession of only a few, when the public was far less ready than now either to make use of the products of that industry or to foresee a great place in the development of American technical art for its factories and its technical staff. We hold the Summer Meeting of 1930, not only for the usual purposes of the presentation of technical papers and the gathering of engineers for social intercourse, but also in honor of the men who formed the S.A.E., who built the automotive industries in their earliest days and who have carried them forward through their most difficult stage. We have with us at this meeting several of that select group who were among the Society's earliest members and are taking the opportunity of turning the clock back, of seeing something of their work and what grew out of it and of its significance."

Nominees for 1931 Announced

The President then invited A. L. Riker, H. W. Alden, E. T. Birdsall, A. C. Schulz, H. G. Chatain, W. G. Wall and P. M. Heldt to come forward to the stage and again assume their place at the helm of the Society.

Saying that while we regard with due reverence the work of the last 25 years, we continue to look forward to the carrying on and further growth in magnitude and importance of that work, Mr. Warner called upon B. J.



FOUNDER MEMBERS OF THE SOCIETY, INCLUDING THREE PAST-PRESIDENTS, AT 25TH ANNIVERSARY SESSION
(Left to Right) W. G. Wall, H. G. Chatain, A. L. Riker, P. M. Heldt, E. T. Birdsall and H. W. Alden

Lemon, as Chairman of the Nominating Committee, to give the report of that Committee and to speak for the several Activities Nominating Committees. Mr. Lemon then presented the names of the nominees for office in 1931, which are given in the accompanying box, saying that he thought the members of the Society would hear with a great deal of gratification the name of the nominee for President—Vincent Bendix,

President Warner then announced that the following telegram had just been received from the White House in Washington, signed by President Hoover:

Please extend my cordial greetings to the convention of the Society of Automotive Engineers and my congratulations upon the twenty-fifth anniversary of the founding of the Society. The technical progress of the art which they profess is one of the marvels of modern times and has contributed incalculably to the liberation of mankind from the limitations of time and space.

HERBERT HOOVER.

Clarkson Elected Honorary Member

Stating that he had another brief announcement to make, President Warner said that this was the first Society activity in many years that Coker Clarkson, whose connection with the Society goes back far beyond the recollection of most of those present, has had to miss and that, although confined by illness at his home, he was with the meeting in spirit, as evidenced by his telegram, elsewhere recorded. He has served the Society for 20 years, continued Mr. Warner, and it is difficult to estimate fairly or put a sufficient value upon the share that his work has had in the Society's growth and in its increasing recognition as holding a leading place among organizations of its kind. Therefore it has been the unanimous desire of all who have been consulted that some special tribute be paid to Mr. Clarkson as befitting the occasion. Two provisions of the Constitution of the Society that had never been used, he said, were Sections 14 and 17, which he read, as follows:

Section 14. Honorary Members shall be nominated by at least 10 members of the Society. The grounds upon which the nomination is made shall be presented to the Council in writing.

Section 17. The election of Honorary Members shall be by vote of the Council taken by letter ballot as provided in the By-Laws. One dissenting vote shall defeat such election.

Last March, 10 Members formally nominated Mr. Clarkson for Honorary Membership and it was a privilege to announce, said Mr. Warner, that without exception every member of the Council voted affirmatively upon the nomination, so that Mr. Clarkson becomes the first and so far the only Honorary Member. Hearty applause greeted this announcement.

Founder Members of the Society Who Are Present Members

Application No.		
2	Henry Ford	Ford Motor Co., Dearborn, Mich.
3	John Wilkinson	Syracuse, N. Y.
4	Edward T. Birdsall	Wright Aeronautical Corp., Paterson, N. J.
9	Herbert W. Alden	Timken-Detroit Axle Co., Detroit
10	Herbert Vanderbeek	Timken Roller Bearing Co., Canton, Ohio
17	Thomas J. Fay	Thomas J. Fay, Inc., Brooklyn, N. Y.
21	Lieut.-Col. Harry A. Knox	Ordnance Department, City of Washington
22	Albert L. Clough	Independent engineer, Manchester, N. H.
24	John G. Perrin	Springfield, Mass.
27	Walter C. Baker	Cleveland
31	Henri G. Chatain	Hammermill Paper Co., Erie, Pa.
32	Charles T. Jeffery	Manufacturer, Philadelphia
33	W. J. P. Moore	New York City
34	David Fergusson	James Cunningham Son & Co., Rochester, N. Y.
35	Joseph Tracy	Manager, New York City
36	W. G. Wall	Consulting engineer, Indianapolis
40	Albert C. Schulz	Bridgeport, Conn.
42	William P. Kennedy	William P. Kennedy Engineering Corp., New York City
45	Arthur J. Moulton	Chateau de LaVerrière, par Mesnil-St. Denis, France
47	Alexander Churchward	Wilson Welder & Metals Co., Hoboken, N. J.

Alvan Macauley's Address by Telephone

Although we think that the Society and the pioneers in it have a certain degree of popular appreciation, it is obviously within the automotive industry itself that that appreciation is keenest, most intelligent and most complete, remarked President Warner. The meeting would therefore be incomplete without some tribute to the pioneers or some comment upon the work of the Society and its significance from a representative of the less directly technical phase of the industry. The gathering was therefore fortunate in having a speaker who represents the industry as a whole and who, although unfortunately unable to be present in person, would address the meeting from his home in Detroit as the representative of the National Automobile Chamber of Commerce—Alvan Macauley, president of the Packard Motor Car Co. Mr. Macauley then delivered his address, which was transmitted by telephone and broadcast in the Convention Hall.

Although he was in Detroit and could not see how many were in attendance, said Mr. Macauley, he knew that it must be many times the number present at the first banquet in 1903 when but 30 sat down and had to listen to speeches. Continuing, he said:

You have grown with the industry and it is well known that the industry has grown in large part because of you. As we look back 25 years, we wonder how the motor-car business ever made the quick start that it did, considering the little it had to work with, the meager experience of its pioneers and the resistance it had to overcome. There was not only relatively little background to guide it in matters technical, but it had to educate men and women, and I might add horses, too, to the motor-car. Good roads had to be built and unfavorable legislation wisely restrained. All of the necessities of the general situation were realized and fortunately a great spirit of cooperation came over all. Especially those in charge of the engineering development came to an early appreciation of the fact that it would be better to work together than to try to progress individually. Without the researches of your technical section and the standardized practices set up, the motor-car industry could not have made the great progress that it has made and it could not have become so soon the Country's greatest industry.

In the few minutes allotted I can do little more than to congratulate you most heartily upon your accomplishments. My great regret is that our own Captain Woolson, who took such a great interest in your Society, cannot be with you tonight. His successes grew out of his experience with you, and he was ever ready to credit you accordingly. We can all mourn his loss not only as a

great engineer and pioneer but also as an executive and salesman, for, like so many of you, he combined all of these qualities. The engineering side of the motor-car business has always had to sell its ideas to those higher up, and the industry has had, and has, many great engineering salesmen.

In conclusion, I can say quite safely that, great as have been your achievements in the past, greater accomplishments are to come. Although so much has been done to perfect the motor-car, it is still not a perfect instrument. None knows that better than you who have blazed the way of its progress. Future improvements may not be sensational in character, but they will be infinitely important, adding to the comfort, safety and all-round more satisfactory performance. Any of you who may be privileged to meet together after another 25 years will find that tonight's impossible has been done and that the transportation world will be still further in debt to that body of men who have always worked together in a spirit of what I may call engineering sportsmanship, regardless of the trademarks or names back of their individual salary checks, which, let us hope, may never grow less.

Riker Presents His Colleagues

As Andrew L. Riker was called upon to take the chair and present those of the founders of the Society who were in attendance at the meeting and seated on the platform, the audience rose and applauded noisily, as it did for each succeeeding speaker. Mr. Riker narrated some of the incidents of the formation of the Society, saying in part:

Twenty-five years ago we had an Association of Licensed Automobile Manufacturers, formed to operate under the Selden patent,

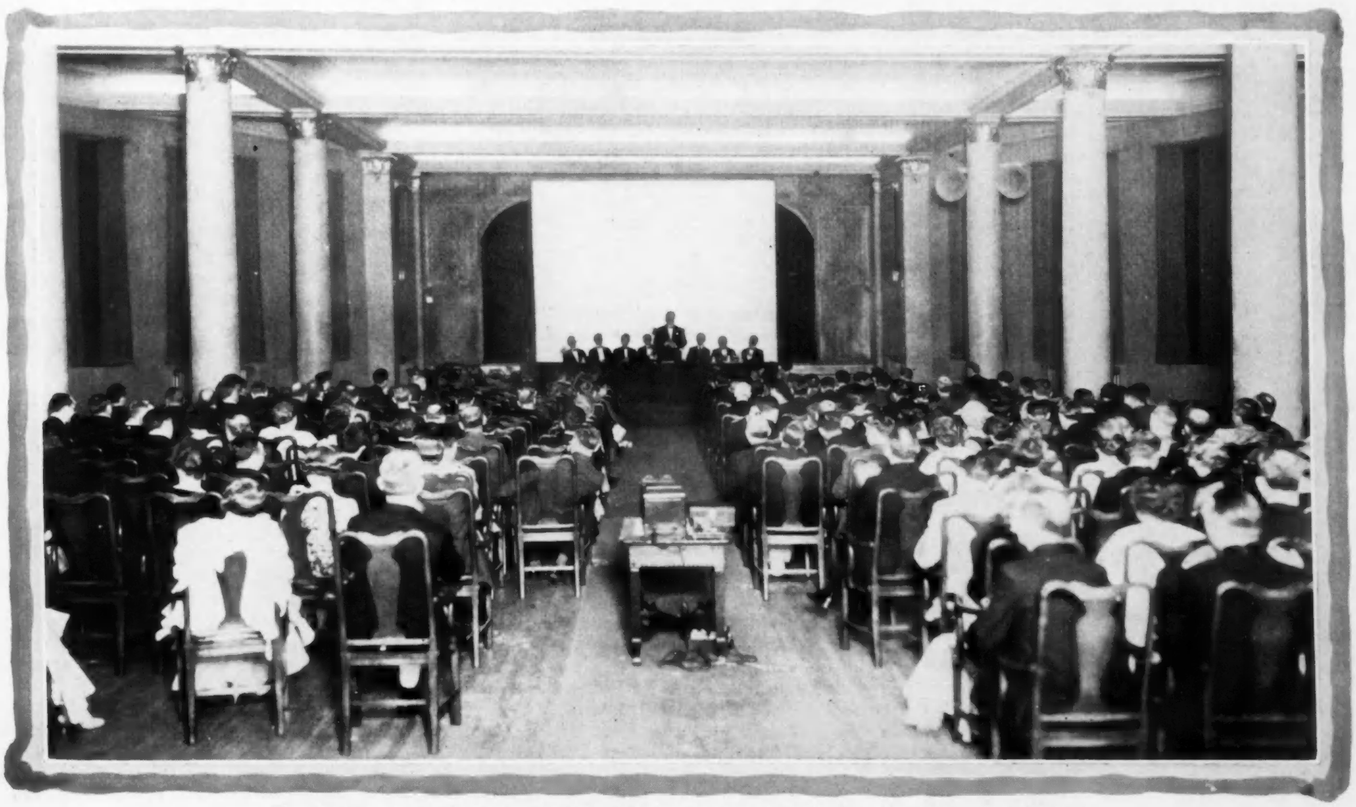
HENRI G. CHATAIN
A Founder Member of the Society

and only a few favorites were permitted to join that association. The A.L.A.M. formed a Mechanical Branch and selected Mr. Clarkson to manage the affairs of that branch, so, as I was chairman of the branch, I have been associated with Mr. Clarkson probably longer than anyone connected with the Society. The purpose of the Mechanical Branch was to have an interchange of ideas between the member engineers of the A.L.A.M., but it was strictly forbidden that anything which went on in that holy of holies was to be transmitted outside. Some of us felt that restriction to be an injustice, and I think that feeling really brought the S.A.E. into being.

Why the founders did me the honor of making me the first President of the S.A.E. I do not know, as it really was difficult for me to work with the chosen and then come out and help those who were trying to get along on the outside. But my sympathy was with those on the outside, because the representatives of the members of the A.L.A.M. came to its meetings with instructions to absorb all the information they could get but not to give out any.

Some of the older engineering societies were very jealous of the S.A.E. The American Society of Mechanical Engineers thought that there was no field for a society of automobile engineers, and the president tried very strongly to induce me to have a merger; in fact, he made me a manager of the A.S.M.E. with the idea, I think, that I would agree to help bring about a merger with the S.A.E. But I said, "We are a body of young men; we have a new industry ahead of us; we have to hew our way. I think the majority of the men in the A.S.M.E. are a little too old to change their ways. I know they have been trying for a great many years to standardize screw-threads and they have not done it yet. We can't wait that long; we must go ahead quickly."

So we struggled along for some time, with the assistance of several of the members, and had meetings now and then. But the Society did not accomplish very much under my management. Mr. Birdsall, who was Secretary and Treasurer, went out and got members and then he voted on them and tried to collect money from them. It was a hard job. I will ask Mr. Birdsall, who was the instigator of the S.A.E., to say a few words.



TWENTY-FIFTH ANNIVERSARY SESSION IN THE MAIN CONVENTION HALL

When Birdsall Started Something

Little did Mr. Birdsall suspect, when he started the movement to form an automobile engineering society, what a giant he was going to be the father of and what it was destined to accomplish. In a humorous address that kept the audience in spasms of laughter, he recounted some of the troubles of his parenthood. He said:

After I made up my mind that, if I didn't start something, somebody else would, I wrote letters to nine of my friends: Mr. Riker, Mr. Maxim, Mr. Alden, Mr. Schultz, Mr. Whiting, Mr. Gibbs, Mr. Vanderbeck, who couldn't back out because he was my cousin, and Henry Ford, who used to race for me on the Empire Track, in New York City. The old 999 that you see downstairs is the car that Mr. Ford used to drive on the Empire Track, and we used to pay him $100 to run around the track a couple of times, letting his hair fly in the air and throwing up clouds of dust on the turns. He used to make the mile in about 58 sec. Barney Oldfield used to go around in the Green Dragon, which was built by the Peerless factory, in a little less time and he always insisted on getting 5 per cent of the gate, so he would go away with $300 or $400 while Henry, not knowing anything about gates, used to get the $100; and I think he spent $35 getting back to Detroit.

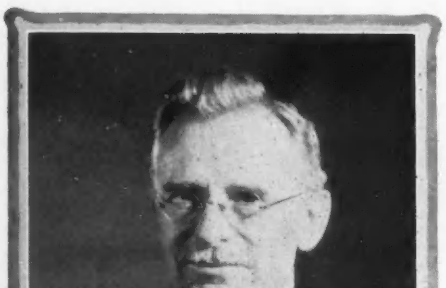
After I wrote the letters to the nine friends that I thought couldn't refuse because I hadn't done anything to them and they had no particular reason, all agreed to join. But those who lived out of town did not attend the meetings and only five came to the meetings at the rooms of the Automobile Club of America, at 753 Fifth Avenue. We had several meetings there and got together a lot of constitutions of other societies and finally wrote a Constitution that we thought was pretty good. We discussed the name of the proposed society and a suggestion was made that we choose something that would include everything. That was voted down and the name The Society of Automobile Engineers was finally selected.

We couldn't find a definition of "automobile" in any dictionary at that time, so we made our own, which was "anything that moves on the earth, under the earth, in the water, on the water, or in the air." As you all know, the name was afterward changed to "automotive," but the word "automotive" at that time—well, we didn't have any motive for making it; in fact, we hardly had the auto.

After we got the Constitution it was a sort of Frankenstein and we didn't know what to do with it, but the next January I interviewed my friend, H. M. Swetland, and E. R. Ingersoll, of *The Horseless Age*, and asked them to back the project. In those days we didn't number our projects; we had only one. They said it was a good thing and they would back it; and Mr. Swetland was 100 per cent. So in January, 1904, in the week of the New York Automobile Show, we held an organization meeting at the Hotel Navarre.

There wasn't much enthusiasm for the next couple of years and it was very hard to pry the $10 a year out of the members. But we didn't have any papers or anything. After the two years Henry Hess was elected Treasurer and I turned over to him $1,277, because we had no expenses; everything was coming in and nothing going out. That was a novel thing in societies; before that I had organized a number of yacht clubs, camera clubs and such things, but none of them had any money; the members wouldn't pay dues. Then we elected a new Secretary and I took a rest.

That is the general history of the start of the S.A.E., and it is extremely gratifying to me that one of my babies grew up. All the rest died of financial insufficiency, but at the present time I think there is no danger of any insufficiency in the S.A.E.; in fact, the Treasurer's report now looks as good as that of some of the investment trusts and other institutions that we look up to.



P. M. HELDT
One of the Founders of the Society and Well Known as a Writer on Automotive Engineering Since the Beginning of the Industry

Nominees for Offices for 1931

For President—Vincent Bendix

For Vice-Presidents—

Dr. G. W. Lewis, Representing Aircraft Engineering

Arthur Nutt, Representing Aircraft-Engine Engineering

W. F. Joachim, Representing Diesel-Engine Engineering

J. A. C. Warner, Representing Passenger-Car Engineering

Carl B. Parsons, Representing Passenger-Car-Body Engineering

A. K. Brumbaugh, Representing Production Engineering

F. K. Glynn, Representing Transportation and Maintenance Engineerig

For Treasurer—C. W. Spicer

For Councilors, to serve through 1931 and 1932—

F. S. Duesenberg, Norman G. Shidle and C. E. Tilston

Broadening of Interests Urged

H. W. Alden next recounted his early experiences when he was associated with Mr. Riker, Percy Maxim, Len Stone and Hayden Eames in the experimental development of an internal-combustion-engined tricycle and a little electric racing car. They were somewhat surprised that the tricycle, which was completed in 1894, ran the first time it was taken out—but not the second time. The racing car was the fastest electric racing car ever built, and Mr. Ford now has it in his museum. It was run in a race at Brookline, Mass., in 1901. In 1895 Mr. Alden built a one-cylinder, vertical-engined car at the plant of the Pope Mfg. Co., in Hartford, Conn., and made the first 500-mile non-stop run in the winter of 1900-1901, just a little less than 30 years ago.

From those days on, continued Mr. Alden, the development of the motorcar has been very rapid and it gives a great deal of satisfaction to look back over the 25 years. The next 25 years is going to be no more interesting, because we have all had a world of fun in those first 25 years of the Society and the 10 preceding years that some of us were in the work. In conclusion Mr. Alden said:

The success of this Society has hinged upon the loyalty and cooperation of the members and the staff, and that is what is going to tell in the future. We are going to have conditions in the future that are a little different; a lot of the problems are solved and we are going to have other than engineering problems. I want to urge upon the Society, particularly the Council, that we reach out a little more into contact with other activities and organizations.

One of the most serious things that confront the automotive industry is the matter of legislation. Perhaps technically this Society ought to keep its fingers off of legislation, but I think it should cooperate a great deal more than it has done. The legislators have their problems, and I have been surprised to find out how much the politician appreciates the constructive assistance you

are giving. I venture to assert that the legislatures of the various States will welcome honest and intelligent cooperation. The most important thing is to help work out problems with them before anything is crystallized. I think the Society can devote a considerable share of its time toward the solving of the general problems of the motor-car industry, cooperating with all the different activities with which it comes into contact.

Chatain and Wall Reminisce

The next speaker was H. G. Chatain, who recalled his investigation in 1897 of the work being done in France by Leon Serpollet with steam vehicles and by Daimler in Germany with gasoline vehicles. He became much interested in the steam vehicles at the start and came home and built a little steam car, doing most of the work himself and paying for it. It burned coke and was started with a blower like a blacksmith's forge. He soon became convinced, however, that the internal-combustion engine was going to supersede anything that would be done in the way of steam apparatus, and turned his attention to motor rail-cars. But he bought for his own use one of the early high-wheeled buggy-type gasoline vehicles built by Charles Duryea. The single-handle control that steered the vehicle by lateral movement, shifted belts for different speed reductions by up and down movement, and more or less regulated the engine speed by twisting the spade-handle, caused an embarrassing catastrophe one day when he took a young lady out riding in Boston. He paid more attention to the charming lady than to the control, and as they were trailing a Brookline street-car, forgot whether to lift the handle or twist it when the street-car stopped. As a consequence, he landed on the

E. T. BIRDSALL AND MRS. BIRDSALL

Mr. Birdsall Drove to the Meeting with Mrs. Birdsall from Hackensack, N. J. On the Return Trip He Drove the 390 Miles from Pittsburgh in One Day

rear platform of the Huntington Avenue car.

What little he has contributed to the automotive industry, said Mr. Chatain, has been along the lines of motor rail-cars. One that he built has accomplished a record, having now run approximately 1,500,000 miles in service. He agreed with Mr. Alden about the importance of looking outside of one's own sphere for inspiration, and mentioned as an illustration that, in the effort to burn oil in internal-combustion engines today, a great deal could be learned by observing what is being done with pulverized coal.

Colonel Wall, the last of the veterans to speak, contrasted the first Summer Meeting of the Society with the present one. Whereas we now have several hundred or a thousand in attendance, the first Summer Meeting, held in Buffalo and Niagara Falls, had only 14. Dave Fergusson was the host and furnished the cars for the ride to the Falls, where the session was held. Others present included Mr. Birdsall, Mr. Riker, Mr. Heldt, Mr. Chatain and Mr. Fay, who later became President. Mr. Fay presented a paper, the subject of which, Mr. Wall said, he could not recall but he thought it probably was along the lines of whether it was not rather complicating an automobile to put as many as four cylinders into it.

At that time Colonel Wall was chief engineer of the National Motor Vehicle Co. and was having a hard time getting the speed of his cars up to 60 m.p.h. He had heard that Mr. Ferguson's cars could make that speed, and mentioned it to Mr. Ferguson, with whom he was riding. The speedometer crept up as the speed increased, to 50 and finally to 60 m.p.h., "and in those days," said Colonel Wall, "speedometers were not used for selling cars, but for telling the speed of the vehicle."

In conclusion, the speaker remarked that the Society has grown wonderfully since those days and has done a great work. It is going to continue to do great work. Under the amended Constitution, which recognizes all the different branches of automotive engineering, there is no reason that it should not continue to be of great benefit to the industry in the future as it has been in the past.

A Sonnet to the S. A. E.

William Herschell, "the bard of Indiana," who was the next and last speaker, recited a number of his poems that touched responsive chords in the hearts of his hearers, as evidenced by the rapt attention and hearty applause accorded to each. Mr. Herschell interspersed many humorous remarks and little anecdotes. His old Scotch dad, he said, put him in the railroad machine-shop in the Indiana town where his father was foreman of the shops. Bill stayed six years and did not know a monkey-wrench from a hammer when he came out. So he decided to do something progressive and became a district superintendent under Eugene V. Debs in the American Railroad Union strike. Then along came President Cleveland's soldiers with fixed bayonets, and Bill went to Canada. He finally journeyed back and started his writing career by working on the *Princeton Engineering News* for $9 a week—if he could get it. He has not been an automobile editor for several years, he said, but he has played the game. He traveled in cars back in the days when "they barked at the road and there was nothing but dust," so he said he had a few sentiments he would like to read. The poem is entitled, The Meaning of the S.A.E., and is printed herewith.

Immediately after the showing of the historic lantern slides, the chairs were cleared from the hall for the Grand Ball.

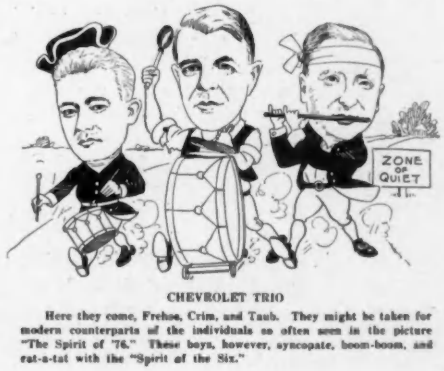
The Meaning of S.A.E.

Twenty-five years of the S.A.E.—
And what has it meant to such as me?
I of the laity, you of the few
God gives only great things to do.
No S.A.E.—and my soul would still
Hunger for faraway vale and hill;
That hallowed shrine, yon distant sea,
Might have, forever, been lost to me!

You that have battled—nights weary long—
A motor gone crazy, a chassis wrong,
Never will know what your toiling meant,
Easing this world of its discontent.
Children go singing your praises now,
Laughter is ringing through field and bough;
Tents at the roadside all seem to be
Joy-castles built by the S.A.E.!

Still, there is something I haven't told,
Proving your hearts to be hearts of gold;
Friendship and fellowship here abide—
Envy is lost and deceit denied!
One that has problems can freely lay
All of his "bugs" in a good friend's way;
Soon they are banished, all care set free—
That is the meaning of S.A.E.!

The "Spirit of the Six"



CHEVROLET TRIO

Here they come, Frehse, Crim, and Taub. They might be taken for modern counterparts of the individuals so often seen in the picture "The Spirit of '76." These boys, however, syncopate, boom-boom, and rat-a-tat with the "Spirit of the Six."

Historical Exhibits and Pageant

Wealth of Early Cars, Engines, Parts and Literature Displayed—Army Vehicles Parade

THE Historical Exhibit, housed in the exposition basement beneath the convention hall, was the center of great interest during the week of the Summer Meeting. Additional exhibits were displayed in the two glass enclosed rooms at the entrance of the hall. In one of these rooms, the United States National Museum and the Smithsonian Institution at Washington exhibited a group of Patent Office models illustrating the steps in the development of the internal-combustion engine from about 1844 to the beginning of the present century. The Smithsonian exhibit included also a group of five early carbureters, four of which were taken from the early cars in the Museum, including those of Charles E. Duryea, the first Haynes, the first Autocar and the first car built by Ransom E. Olds. This series also included a carbureter built by A. L. Dyke in 1901, which was probably the first carbureter built for sale in the United States. The Patent Office model of the George Selden road carriage of 1879 was also displayed, together with the specifications of the original patent.

Manly and Wright Engines Displayed

In the other room, the beginning of the airplane-engine development was shown by the original Manly engine of 1901, loaned by the Smithsonian Institution, and the engine that was handbuilt by Wilbur and Orville Wright and used in their early experimental flights in 1903 and 1904, being loaned by General Motors Corp. The Manly engine is of particular interest in that it contains so many features of modern aircraft-engine practice, being in this respect many years ahead of its time. It is a five-cylinder air-cooled radial engine that developed 52 hp. at a speed of 950 r.p.m. and weighs only 115 lb.

The carbureter used with the Manly engine was also displayed. It is a surface carbureter consisting of a large copper tank filled with small pieces of tupelo wood over which the fuel was permitted to drip and through which the air was drawn to vaporize the fuel. The Museum also exhibited photographs of 11 of its collection of early automobiles, starting with the Charles Duryea car of 1892.

The Wright engine is a four-cylinder valve-in-head model with a flywheel. This engine developed 30 to 35 hp. and weighs only 180 lb.

The Packard Motor Car Co. exhibited the original Packard Diesel aircraft-engine, designed by the late Capt. L. M. Woolson. This, the first Diesel aircraft-engine in the United States, made its initial flight on Sept. 19, 1928. It later flew from Detroit to Langley Field, Va., 650 miles, on May 13, 1929, with a total fuel cost of $4.68. This is a nine-cylinder air-cooled radial engine equipped with crankshaft torque shock-absorbers of rubber.

Collection of Historical Photographs

On the walls of the room where the engines were displayed were 50 historical photographs shown by Nathan Lazarnick, the photographer. The first Vanderbilt Cup, won by J. H. McDuffee and loaned by him to the exhibit, occupied a prominent position on a table in the center. In a case nearby were shown the Society's Wright Brothers and Manly Medals.

In the hallway between the rooms was the Locomobile used by General Pershing at the front in the World War, and in the reception room a collection of photographs of Past-Presidents of the Society. The National Automobile Chamber of Commerce loaned a complete set of Handbooks of the Automobile to the Historical Exhibit, and these were prominently displayed in the upper hall at the entrance to the convention hall. In the same room there was a display of S.A.E. Standards, S.A.E. JOURNALS and other interesting S.A.E. material.

Early Motor-Cars Exhibited

Descending into the basement, the first sight was a 1902 Peerless touring-car, with its rear-entrance tonneau, tallyho basket, beveled plate-glass windshield and characteristic type of baggage rack. Immediately beyond were scores of small exhibits prepared by the several technical journals, newspapers and magazines, depicting early advertising and editorial material, the first volumes of early trade papers, books, pamphlets, catalogs, instruction books and other interesting literature of 25 years ago.

The Timken Axle and Timken Roller Bearing companies contributed interesting display boards portraying the progress of the last quarter century in their products.

The B. F. Goodrich Rubber Co. display included samples of tires and tire construction showing the progress from 1898 to date; also an interesting group of early photographs of tours and races, including a visit of the Society to the Goodrich plant in 1912.

Immediately beyond, the Goodyear Tire & Rubber Co. showed a 10-ft. model of the new 6,500,000-cu. ft. 75-passenger Zeppelins which the Goodyear-Zeppelin Corp. is building; also models of its semi-rigid airships and sections of duralumin construction.

A glass case contained the exhibit of the Champion Spark Plug Co., comprising different spark-plugs, totaling 80 spark-plugs, mostly cut away, ranging in dates from 1907 to 1930.

The exhibit sent by the Delco-Remy Corp. comprised 40 specimens of starting, lighting and ignition units, ranging from 1895 to date.

A glass-enclosed display of Eclipse-Bendix drives showed a Bendix drive of 1913 and one of present-day design.

Two cutaway Spicer universal-joints formed the exhibit of the Spicer Mfg. Co.

Ford Racing Car and Riker Electric

Occupying a prominent place in the aisle was the famous 999 racer, built and loaned by Henry Ford. This racer was the first car to break the mile-a-minute record, being driven by Henry Ford himself in January, 1904, at Baltimore Bay, Mich. The time was 39 2/5 sec. on a cinder track on the ice. The four-cylinder engine, with 7 x 7-in. bore and stroke, developed 80 hp. It is directly driven through bevel gears on the rear axle. The racer was named after the famous 999 New York Central locomotive, which was the fastest thing on wheels at the time.

Alongside the racer stood a three-wheeled electric vehicle designed and built in 1898 by the first President of the Society, Andrew L. Riker. This was used as Mr. Riker's personal car at Stamford, Conn., principally as a station wagon. Mr. Riker presented the car to the Ford Museum, and it was loaned to the Exhibit by Mr. Ford.

A 1903 Model-A Ford touring car and the 15-millionth Model-T Ford came next. The former, a phaeton with a rear door, with a two-cylinder opposed engine having 4 x 4-in. bore and stroke, was loaned by Mr. Ford, and the latter was loaned by the Ford Motor Co. Over in the far corner of the exhibition hall were a Holsman horseless carriage and a 1903 Oldsmobile.

The Holsman Motor Buggy

The Holsman was one of the most remarkable specimens exhibited. Built in 1902, it sold for $800. It was equipped with a rope drive and a two-cylinder opposed air-cooled engine. Lighting was furnished by a carbide lamp in the center of the dashboard in front of the driver. The car originally belonged to a minister who made calls through the western part of Ohio in this motorized buggy. The car was presented to the Ford collection by C. E. Williams, Secretary of the Miami County Automobile Club, Piqua, Ohio. The Oldsmobile, a 1903 model with curved dashboard, excited much comment from visitors whose memories were of sufficient length to enable them to have reminiscences of the days when this ancient model was strictly modern.

NOTABLE OLD-TIME CARS SEEN IN THE HISTORICAL EXHIBITION AND IN THE PAGEANT

(1) Electric Tricycle Built by Andrew L. Riker in 1898. (2) Steam Car Built by Howard E. Coffin in 1899. (3) First Franklin Car Sold, Model of 1902. (4) Renault Racing Car That Won the Paris-Bordeaux Race in 1901. (5) Holsman Rope-Drive Buggy, Built in 1902. (6) First Model Three-Cylinder Two-Cycle Paige, Built in 1909. (7) Vertical-Engine Peerless Car, with Top and Windshield, Built in 1902. (8) Round-the-World Hupmobile, Built in 1910, That Visited 26 Countries. (9) White Steam Car of 1905 That Was Driven from Cleveland to the Summer Meeting under Its Own Power. (10) Racing Car Built by Henry Ford in 1904 and Driven by Him in Track Meets. (11) Locomobile Limousine Used by General Pershing at the Front in the World War. (12) Rear-Entrance 1903 Model-A Ford Touring Car That Preceded the Ubiquitous Model T

The original Chrysler car, built in 1924, and Dodge car No. 374, built in 1915, came next. These were flanked with a voluminous display of Chrysler and Dodge photographs and early catalog material.

The Edward G. Budd Mfg. Co., of Philadelphia, sent one of its early touring-car steel bodies, built in 1915, and an example of its modern monopiece construction, built in 1929. Both of these bodies were sectioned so that the condition of the body and the upholstery was clearly shown. Around the wall near this exhibit were 12 photographs portraying old-time electric cars, parts, trucks and buses, designed by Joseph Ledwinka, chief engineer of the Budd company, and loaned by him to the Historical Exhibit.

The Ross Gear & Tool Co. display included five examples of early and modern steering-gear construction.

On the other side of the hall were shown a 1904 model Reo car with a rumble seat. An accompanying exhibit comprised two transmissions, one being an old planetary transmission similar to that in the 1904 car exhibited and the other being the present model, and a three-part folding stand containing an illustrated statement of engineering features fostered by Reo since 1905.

The White Steamer of 1905

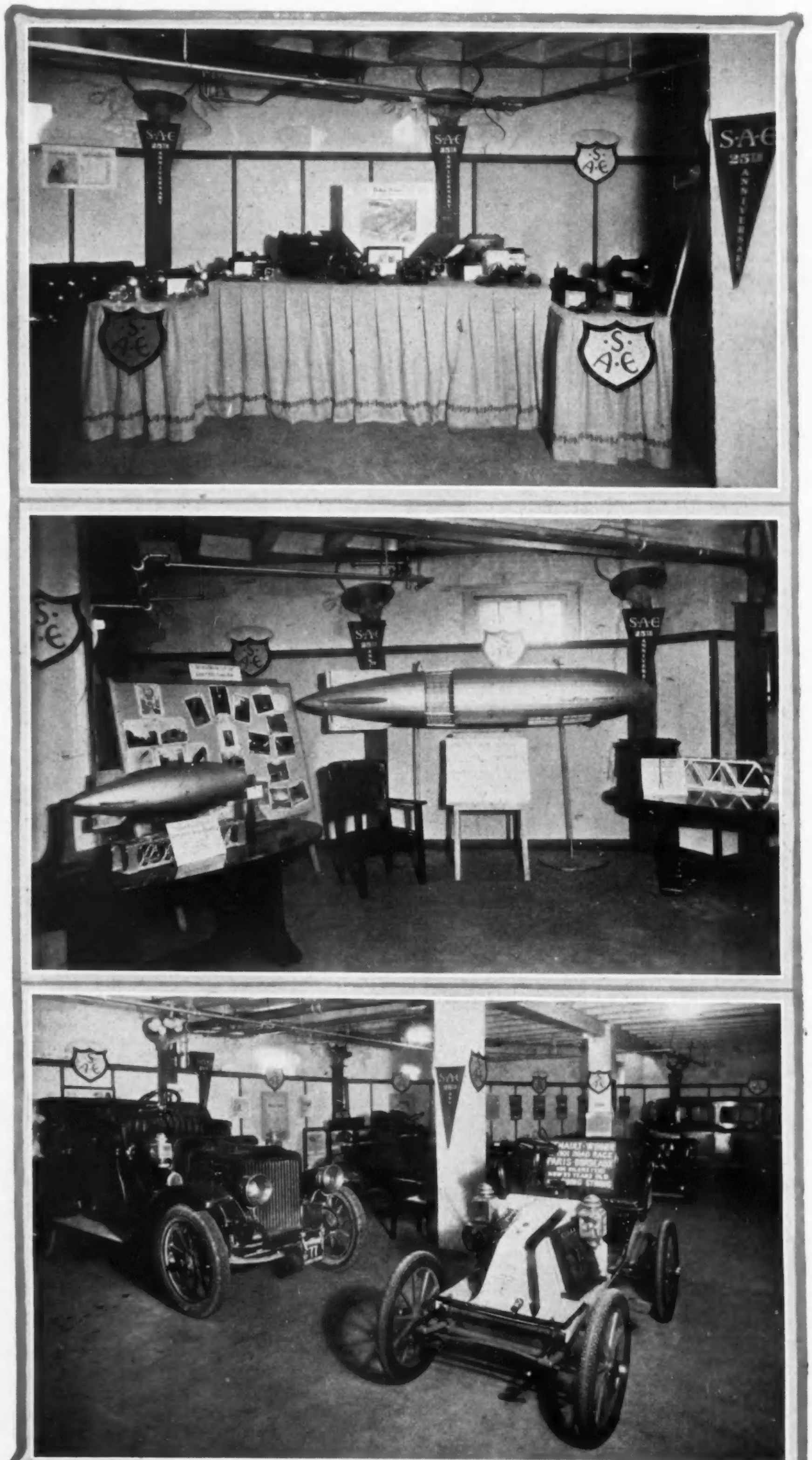
Next to the foregoing exhibit was a White steamer, a touring-car built in 1905, which was driven to French Lick from Cleveland on Monday of the Summer Meeting week under its own power.

Two Hupmobiles were shown, one being a runabout, Model 20, No. 176, built in 1909 and believed to be the oldest Hupmobile still running. The other was a round-the-world car, built in 1910, which visited 26 countries in two years and was the first automobile seen in 14 countries.

In front of the cars just mentioned was the Renault runabout with which Louis Renault won the road race from Paris to Bordeaux in 1901, and in the aisle beyond was a two-cylinder steamer made in 1899 by Howard E. Coffin, Past-President of the Society, while he was an engineering student at the University of Michigan.

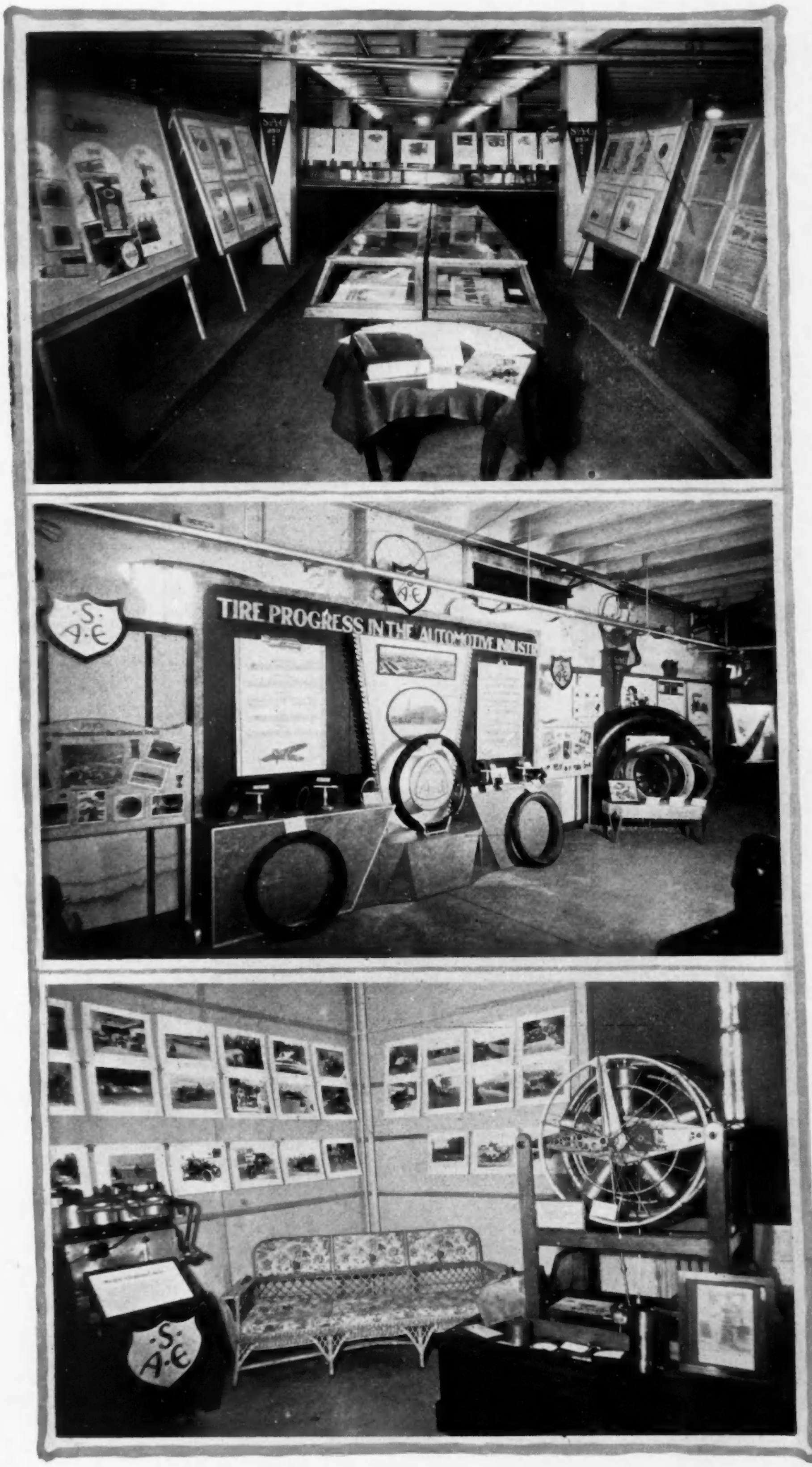
A three-cylinder two-cycle 20-hp. Paige runabout with top and a 1902 Franklin four-cylinder air-cooled car came next. The former was the first-model Paige built in production in 1909. The latter was the first Franklin car sold, the price being $1,200.

Other visiting vehicles, not displayed in the hall, were the Studebaker President Eight stock-car which had run 30,000 miles in 26,326 consecutive minutes; the Diesel-electric motorcoach of the Public Service Coordinated Transport, the Robert Bosch Magneto Co.'s Diesel-engine truck and the Cummins Diesel-engine car.



GENERAL VIEWS OF THE HISTORICAL EXHIBITS

(Top) Display of Progressive Development of Automotive Electrical Apparatus. (Center) Models of Projected American Airships. (Bottom) Display of American and French Cars of 25 and 30 Years Ago



IN THE BASEMENT OF THE MAIN CONVENTION HALL

(Top) Early Periodical Literature Dealing with the Automobile. (Center) Display Showing Progressive Tire Development. (Bottom) First American Airplane Engines and Photographs of Early Automobiles and Contests

Aeronautic Exhibits

The aeronautic exhibit contributed by the Curtiss-Wright organization contained one of the three original Wright Whirlwind engines used by Commander Richard E. Byrd, Jr., in his flight over the North Pole; the two-cylinder opposed aeronautic engine designed and built by Charles L. Lawrance; the original Curtiss eight-cylinder air-cooled motorcycle, designed, built and driven by Glenn H. Curtiss in 1907, with which he established a speed record of 137 m.p.h., which has never been broken; a cutaway electric-driven six-cylinder radial airplane Challenger engine; a Challenger cylinder, piston, wristpin, intake and exhaust rocker-arms; and a 12-cylinder, 600-hp. Conqueror engine.

On one of the tables in the center of the hall was an extremely interesting personal exhibit of Charles, Ralph and Dan Teetor, of the Perfect Circle Co. Here were shown a number of early small-sized steam and internal-combustion engines, and some quaint and interesting photographs of the boys and their early work.

Government Production Charts

The United States Bureau of Foreign and Domestic Commerce prepared five production charts for the Historical Exhibit, showing the United States production and exports (*a*) of motor-vehicles, passenger-cars and trucks from 1907 to 1929; (*b*) of motorcycles, showing production, exports and exports-value from 1914 to 1929; (*c*) of airplanes from 1919 to 1929; (*d*) of motorcoaches from 1916 to 1929; (*e*) of marine engines, including Diesel and semi-Diesel engines from 1914 to 1929.

The Curtis Publishing Co. prepared for the exhibit a large number of early and contemporary advertisements of the leaders in the industry. Notable among these was the first two-color full-page advertisement of an automobile ever run in the *Saturday Evening Post;* namely, an advertisement of the Oldsmobile, quoting Mother Shipton's prophecy: "Carriages shall without horses go." The first truck advertisement ever carried by that periodical was also exhibited. One of the early volumes of the *Saturday Evening Post* was on display, containing a number of small pieces of copy, setting forth the modest claims of the pioneers of the industry. Side by side with these was one of the early advertisements of the French Lick Springs Hotel before it had attained its present size and popularity.

The *New York Times*, the *Sun*, the *Herald-Tribune*, the *Detroit Free Press*, the *Denver News-Times*, the *Cleveland Leader*, the *Cincinnati Enquirer*, the *Chicago Daily News*, the *Indianapolis Morning Star* and the *St. Louis Globe-Democrat* sent interesting examples of early advertising and editorial pages,

as did also *The Woman's Home Companion, Collier's, The American Magazine, Nation's Business, Harper's Bazar, Scribner's*, and *Life*. There were hundreds of photographs, early books, copies of early automotive and engineering journals, as well as a comprehensive collection of material concerning historical vehicles of the 18th and 19th centuries, loaned by the Public Library of Newark, N. J.

In the opinion of A. L. Riker and Peter M. Heldt, both of whom have been familiar with the industry since its inception, the collection represented the greatest assemblage of such material that has been brought together in the history of the industry and came as a fitting climax to the quarter-century celebration of the Society.

Thursday's Historical Pageant

On Thursday afternoon, at 2:30, the historical pageant proceeded past the reviewing stand, which was the veranda of the hotel. This pageant was made up of a number of cars from the Camp Holabird caravan, combined with the antique cars withdrawn from the Historical Exhibit in the convention hall basement. Under the direction of Col. Stayer and Major Lawes, the Camp Holabird caravan had set up its headquarters in Dry Hollow, a picturesque ravine just beyond the golf course. The outfit was made up of a number of very interesting types of Army vehicles.

The headquarters company equipment consisted of one experimental Viking sedan, one Ford touring car, one Dodge touring car, one Dodge cross-country car, one Harley-Davidson solo motorcycle and one six-wheel gas-electric truck. Whenever the convoy is in camp, the gas-electric truck supplies the necessary lighting for the camp.

The first company equipment consisted of one Chevrolet touring car, four U.S.A. Class B three-ton cargo trucks and one U.S.A. Class B tank truck. These Class B trucks are the same type that were produced for the Army during the war and were designed by a board of engineers, many of the members of which attended the Summer Meeting this year. The Society was the directing spirit behind the development of this vehicle for the Army.

In addition to the Class B trucks, the first company also had one White reconnaissance car, one experimental 1½-ton Willys-Knight cargo truck and one experimental Willys-Knight ¾-ton cargo truck, one Indiana 1½-ton cargo truck, one six-wheel Class B tank carrier of the same type that is used for the highway transportation of body tanks, one Coleman 1½-ton four-wheel-drive cargo truck and one Coleman 3-ton four-wheel-drive cargo truck.

The second company was equipped with one Chevrolet touring, Company 4 Class B tank truck, one White reconnaissance truck, one experimental heavy ambulance of a new type, fitted upon the new A-2 cargo chassis, one experimental 1½-ton Selden cargo truck, one experimental 3-ton four-wheel-drive Oshkosh cargo truck, and one type-A3 heavy office-truck mounted upon the first Class-B six-wheel chassis which the Army developed. This same vehicle has just attended its third S.A.E. Summer Meeting, having been first inspected by members of the Society at the Summer Meeting at Spring Lake, N. J., in 1923.

In addition to the aforementioned vehicles, the convoy had one six-wheel Class-B tank carrier which was the pilot model for this type of vehicle in the Army; also one six-wheel horse-van which has been developed for the purpose of highway transportation of officers' mounts.

The purpose of the movement of this convoy of vehicles across the Country is the training of convoy officers in the handling of a large body of vehicles under highway-transportation conditions as they are found today. The organization left Camp Holabird on May 5 and proceeded to Detroit, where inspection visits were made to many of the prominent factories. It then drove from Detroit to French Lick for the purpose of enabling the Society members to inspect the new types of vehicle used in the Army and, after the close of the Summer Meeting, proceeded to Jeffersonville, Ind., en route to its home station at Camp Holabird, Baltimore.

Field Day Furnishes Fun

Novelty Races by Men and Women, Turkish Pookah and Egg-Throwing Create Hilarity

ALTHOUGH the flag-bedecked canvas roof, the peanuts and the pink lemonade were absent, the circus atmosphere certainly pervaded the Mystery Field Day held Tuesday afternoon on the lower golf course of the hotel. Several hundred members and ladies gathered under the trees to watch and participate in the 16 competition events listed on the handbills distributed. The strains of the hotel orchestra, the stands of prizes and paraphernalia for use in the contests, and the rows of chairs set up on the terrace along the path bordering the golf course lent a gala atmopshere to the occasion. The Society's loud speakers were set up so that all could hear the announcements.

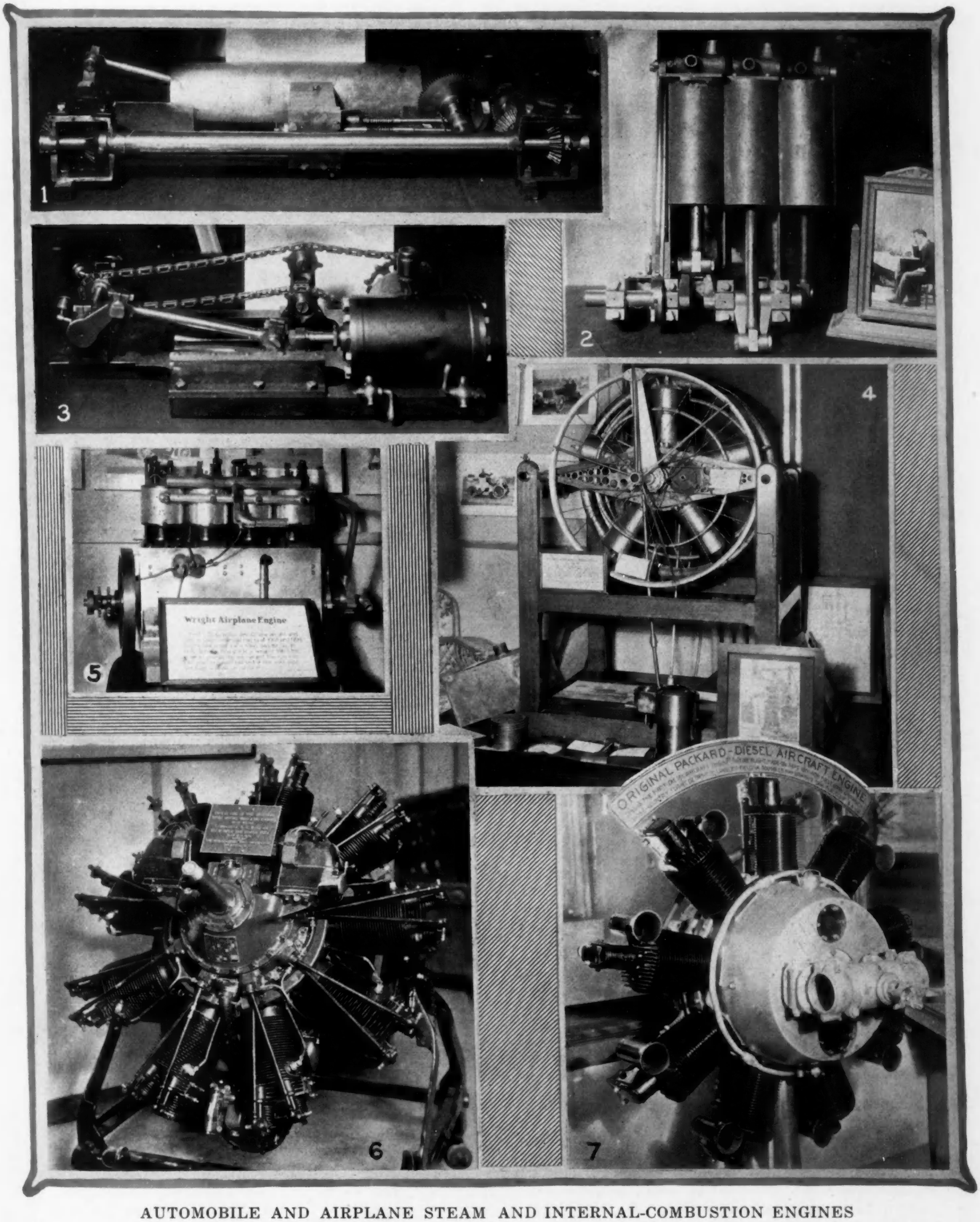
Ringmaster S. S. Dickey, of Saranac Inn fame, with his megaphone and his come-on encouragement, had little difficulty in securing volunteers for the various events. There were no serious, strenuous contests to strain unaccustomed muscles or overtax weak hearts, and the prizes had been selected with a view to fun instead of worth, for the whole plan was to make the occasion one of pure amusement in which any man or woman could enter in the spirit of enjoyment and of furnishing laughable entertainment for others. The ladies who took part freely were pronounced "good sports" for their willingness to appear ridiculous so that the onlookers might have a good laugh. From start to finish of the events, which were run off in quick succession, the Mystery Field Day was a pronounced success. The weather was perfect, the setting ideal and everybody in the best of humor. They could not well be otherwise, under the circumstances and conditions.

Ludicrous and Skillful Displays

Every event was in the nature of a novelty and it would be hard to say which were the more ludicrous; probably the consensus of opinion would award the laurels in this regard to the Volstead race, while the egg-throwing contest for men undoubtedly required the greatest amount of skill and dexterity.

A number of men and women taking part in the Volsted race stood in two lines facing each other as partners and were directed to place one end of a stick a yard long on the ground in front of them, rest their hands on the upper end of the stick and their foreheads on their hands with their eyes shut. Then they walked around the stick six times, picked up the stick and ran to their partners, who received the sticks, performed the same evolution and ran—or attempted to run—to the other line. The results, as the contestants staggered between the lines and often fell down, plainly revealed why this event got its name.

In the egg-throwing contest men teammates formed in two opposed lines close together and the men in one line were given a fresh hen's egg apiece. These were tossed to their teammates and back again, then each line stepped backward one step and repeated the operation. Each time an egg was broken the team responsible for the catastrophe dropped out. Finally, only two teams remained, and these were separated by a distance of nearly if not



AUTOMOBILE AND AIRPLANE STEAM AND INTERNAL-COMBUSTION ENGINES

(1) Rotary-Valve Steam Engine Built by Charles Teetor in 1899 and Used in an Inspection Car. (2) Three-Cylinder Teetor Steam Engine of 1896 to 1898, of the Type Which Played an Important Part in Automobile Development. (3) Opposed-Type Gasoline Engine Design in 1896 and 1897 by Charles N. Teetor for Use in Light Railway Inspection Cars. (4) Manly Radial Engine Used in Prof. S. P. Langley's Experimental Flying-Machine. (5) Four-Cylinder Gasoline Engine Hand-Built by Wilbur and Orville Wright and Used in Their History-Making Flights in 1903 and 1904. (6) One of the Three Wright Whirlwind Engines with Which Commander Richard E. Byrd, Jr., Flew Over the North Pole in 1926. (7) First Packard Diesel Aircraft-Engine, with Which the Initial Long-Distance Flight from Detroit to Langley Field, Va., Was Made in 1929

quite 100 feet. Then in the last toss both teams broke their egg, resulting in a tie.

Passing the Match-Box

The match-box-passing relay race for men and women was another contest that evoked continuous shouts of laughter as the passers endeavored to affix the boxes from the nose of one person in line to that of the next. The crab race, in which a line of men with paper bags drawn over their heads progressed backward toward a goal on their hands and feet with their backs to the ground, showed some contestants traveling in wide circles farther from the opposite side than when they started.

Much amusement was provided also by the Jiggs and Maggie derby, which was a competition in rolling-pin throwing by women and was indicative of the degree of danger incurred in a domestic argument by the married men and the benedicts contemplating embarking upon the matrimonial sea.

The heralded Turkish Pookah turned out to be a pea-shooting contest with bean-blowers by a line of men seated Turkish fashion about a dozen feet from a row of graniteware pans, into one of which each man tried to shoot as many peas as possible in 3 min. The tintinabulation of the peas upon the pans raised high hopes of a goodly mess of legumes, but the total count of the prize-winner was only 24.

Some of the men were threatened with apoplexy in the bag and balloon race for men and women, in which the women of the contesting teams blew up and burst paper bags and the men inflated rubber balloons until they burst the balloons or a blood vessel. This was the only event besides the Jiggs and Maggie derby in which much danger was involved.

Army Encampment Visited

Everything combined to make the whole affair colorful, hilarious and most enjoyable. At the conclusion, the contestants and spectators were urged through the loud speakers to visit the encampment of the Motor Transport Convoy from Camp Holabird, Md., which was within sight beyond the golf courses to the west. This motorized military detachment had driven out from Camp Holabird expressly as a feature of the Summer Meeting and to celebrate the Society's 25th anniversary, and most of the Field Day group were glad to troop over to the encampment afoot to inspect it and to show their appreciation.

The encampment included a score of motor-trucks and some 15 Army tents. Among the trucks were a six-wheel van for transporting half a dozen horses, a heavy-duty Army hospital truck, an officer's office in a truck, a six-wheel truck with a cleated tail-gate ramp for moving a gun or a searchlight mounted on wheels, a tank truck for the gasoline and oil supply, and various six-wheel, four-wheel-drive and more conventional transport trucks, and an Army kitchen built on a truck. While the spectators were present the military ceremony of striking the colors was given.

On the way to and from the encampment, the visitors also inspected several airplanes that used the golf course

SCENES THAT MEMBERS ATTENDING THE 1930 FRENCH LICK MEETING WILL NOT FORGET

(Upper Left) Encampment of Military Train from Quartermaster General's Department at Camp Holabird, Md. (Upper Right) Arriving at the Hotel from the Special Train from New York City. (Lower Left) Disembarking from the Special Train. (Lower Right) Resting from Technicalities and Sports on the Shady Lawn

SOME OF THE MYSTERY FIELD DAY SPECTATORS AND EVENTS

as a landing-field during the Summer Meeting.

Here follows the list of the Mystery Field Day events and the winner or members of the winning team in each contest:

EVENT	WINNERS	
(1) Crab Race	C. S. Bruce	
(2) One Out	C. S. Bruce	
(3) Match-Box Passing	W. E. England	Mrs. L. A. Chaminade
	James T. Greenlee	Janet Rockwell
	Walter F. Rockwell	Elizabeth Warner
(4) Bag and Balloon Race	J. Eisinger	Mrs. J. Eisinger
	Jule Marshall	Mrs. J. T. Greenlee
	Frank A. Taylor	Genevieve Walmsley
(5) Elephant Walk	H. E. Blasingham	M. H. Landis
	R. W. Brown	J. W. Tierney
	B. W. Keese	T. H. Wickenden
(6) Volstead Race	Walter F. Rockwell	Janet Rockwell
(7) Stepping Stones	John Eisinger	Mrs. J. Eisinger
(8) Jiggs and Maggie Derby	Mrs. G. Edgar	
(9) Pipe Race	J. W. Tierney	
(10) Kangaroo Balloon Race	Genevieve Walmsley	
(11) Sack Relay	R. Baggaley	J. A. Day
	C. S. Bruce	D. B. Grasett
	R. M. Curts	Jule Marshall
(12) Cigar-Smoking Race	M. H. Landis	
(13) Balloon-Tail Race	R. M. Curts	
(14) Jump the Spot	Elizabeth Warner	
(15) Turkish Pookah	T. H. Wickenden	
(16) Egg-Throwing Contest	G. J. Bundy	D. B. Grasett

A Movie Voyage with Kettering

The General-Session Audience Visits the Panama Canal, Cocos and the Galapagos Islands

ON MOTION of W. R. Strickland, duly seconded and unanimously carried, the following telegram was sent to Coker F. Clarkson at the opening General Session held during the evening, May 25.

The Society of Automotive Engineers assembled here, missing your genial presence, send you their heartiest greetings and sincere good wishes for a speedy and full recovery.

A lengthy illness and somewhat slow convalescence prevented Mr. Clarkson's attendance at this 1930 Semi-Annual Meeting of the Society, which was a source of great regret to the more than 700 members and guests present.

Chairman J. A. C. Warner then called for the showing of lantern-slide caricatures of Charles F. Kettering, the principal speaker, and commented upon them humorously as a part of his introduction. He said also that "Mr. Kettering not only preaches the gospel that the desire for luxuries and the creation of a demand for luxuries both go hand in hand with prosperity; he practises it. He has bought a yacht. This yacht of his and his experiences at the Galapagos Islands form a basis for his address tonight." Thereupon, two small negro boys dressed in tropic costumes came the entire length of the main aisle, up some steps and onto the platform, where they were each presented with a box of candy. Continuing, Chairman Warner announced that these messengers from the Galapagos Islands had brought the following message: "Give our respects to your speaker and remind him that he once said that it is always best to tell the truth because you do not then need to remember what you might have said previously." Thereupon, Mr. Kettering came to the rostrum while the audience arose and applauded.

Novel Features of the Diesel-Engined Yacht

Mr. Kettering said in part that some of the interesting features of the yacht included a somewhat different type of Diesel engine, principally from the viewpoint of injection and lubrication.

In ordering the engines, only two specifications were given; namely, there was to be no smoke at full load and no brass oil-cans in the engine room. These seem foolish, but they turned out to be somewhat more technically desirable than they appeared superficially. As for the injection system, the pump was built into the nozzles, the object being to overcome certain difficulties experienced with ordinary nozzles. This design caused much trouble; it was the first that they had made, but nothing serious developed. Injection pressures, depending upon the viscosity of the oil, ran up to be from 5000 to 7000 lb. per sq. in.; they broke many of the low-pressure pipes, he remarked. Those who assembled these Diesel engines had never before assembled low-pressure piping; they used high-pressure-pipe wrenches and broke nearly all of the brass fittings.

Specially Equipped Powerplant

Continuing, the speaker remarked that alternating-current generators were used in place of flywheels as synchronizing devices to tie the two engines together; this made them operate at exactly the same speed and at exactly any desired crankshaft-phase relationship. The two six-cylinder engines were each of 500 hp. at 350 r.p.m., and represented roughly 80 hp. per cylinder at cruising speed. They used 80-kw. generators so that, if one cylinder went out of service on one engine, they could still carry on and have some power to spare, he said.

The best arrangement they found, Mr. Kettering continued, was to tie the two engines together as one 12-cylinder engine. Due to their great weight, all the vibration is eliminated. Those who know say that the yacht has less vibra-

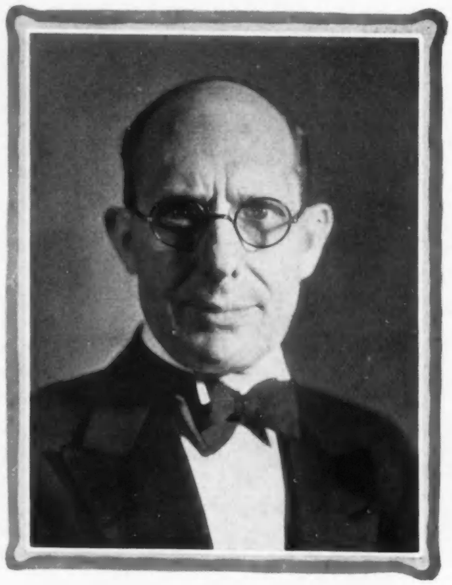
RACES AND OTHER FIELD DAY EVENTS THAT CREATED GREAT HILARITY

tion than that of an electrically driven ship and that the two engines of the yacht produce very much less noise. Load meters were installed between the engines to indicate the wattage passing from one engine to the other; they are equivalent to three-phase wattage-relays.

Other features installed were, according to Mr. Kettering, three-phase induction motor-generator sets between the alternators and the storage batteries; and a Sperry stabilizer whose wheel is about 5 ft. in diameter, weighs 5 tons and which runs at a speed of 1800 r.p.m. When the stabilizer is connected to a synchronous set of Diesel engines such as this, it acts as an additional flywheel to the Diesel engines and steadies their operation very greatly. These and other engineering features described by Mr. Kettering proved most interesting to the audience which, thereupon, figuratively boarded the yacht and cast off.

Features of the Panama Canal

Motion pictures of the Panama Canal were shown and Mr. Kettering commented upon them and upon conditions relating to how problems of sanitation have been solved such as the almost complete elimination of yellow fever and malaria. He said that the trip in his yacht began at the Canal and that the first course extended to Cocos Island, which is about 13 miles in circumference. It is a verdant island with luxuriant vegetation and has two good harbors. The Pacific pirates used to come there, careen their ships on the beach and bury their treasures, he said.



PAST-PRESIDENT C. F. KETTERING
Who Described and Showed Motion Pictures of the Trip to the Galapagos Islands in His Diesel-Engined Yacht at the General Session

Course Set for the Galapagos

From Cocos Island, the course was set for the Galapagos Islands, an archipelago on the equator which is about 125 miles in diameter, being composed of about 14 major islands and several hundred smaller islands, all entirely volcanic in origin. This, said Mr. Kettering, is the newest land on earth and has the oldest biological specimens. Fresh water exists on only one of the islands. The others are composed of nothing but lava. The party spent about two weeks there and then came back to the Panama Canal and made various side trips, one being a trip up the Chagres River.

The movies showing the experiences of the party during their visit were wonderfully unique and most intensely interesting. Mr. Kettering's comments upon them were vividly descriptive and had a personal appeal that held his audience spellbound. The scenes that exhibited the animal and bird life of the Islands were splendidly instructive and thoroughly spellbinding. To see the marine iguanas, the seals, the penguins, the flightless cormorants, the land crabs, the turtles, the sharks and the birds, in action in their native habitat, was like actually being there in person.

Photographs at the Meeting

NEXT to President Warner, John Warner and the S.A.E. staff members who attended to all arrangements for the meeting and conducted the sessions and sports and social events, probably the busiest man at the Summer Meeting was Will Krohn, of Nathan Lazarnick's Cleveland office, the S.A.E. official photographer. Between photographing all the historical exhibits in the basement of the Main Convention Hall, making portraits of the chairmen, speakers and principal discussers at all the sessions and tak-

VISITING DELEGATION FROM THE FRENCH SOCIETY OF AUTOMOTIVE ENGINEERS

ing action pictures of the Mystery Field Day events, the Pageant and the Army encampment, Mr. Krohn found it necessary to be ubiquitous and ambidextrous.

Mr. Lazarnick was prevented from attending the meeting in person and assisting in the work because of the illness of Mrs. Lazarnick. His display of enlargements of many photographs of early American automobile competitions was a pleasing feature of the historical exhibition that recalled vividly to the memory of old-timers the thrilling days of the Vanderbilt Cup Race, the Climb to the Clouds, the Pittsburgh Endurance Run and similar events of the early years of the century. Most of the Summer Meeting photographs are "by Lazarnick."

Standards Committee Session

Proposed Additions and Changes in Standards Are Passed After Argument

MEMORIAL DAY, cutting short a week that was so full of technical sessions, anniversary exercises and sports, made it necessary to hold the meeting of the Standards Committee on Sunday afternoon, May 25. Members of the committee and others, to the number of about 45, were in attendance when Chairman Arthur Boor rapped the gavel to open the meeting, and he was called upon to make exact parliamentary decisions before it closed.

The meeting started off tamely enough with the report of the Aircraft Division, designated by Chairman Boor as the youngest—but very active. J. F. Hardecker, chairman of the Division, presented several changes in the report as printed in Section 2 of the S.A.E. JOURNAL for May. Most of the changes had to do with tolerances on dimensions and some were to secure agreement with Army and Navy standards. With the corrections, the proposals were passed.

Upon telegraphic request from Chairman Arthur Nutt, of the Aircraft-Engine Division, action on the proposals of that Division in regard to dimensions for No. 50 hub and for propeller cones and nuts was deferred and they were referred back for further consideration. The No. 0 tapered shaft-end proposed was reported for Recommended Practice rather than Standard, and the present No. 1 and No. 2 sizes were recommended to be given the same status.

Changes in the tables of angular and open or separable-type ball-bearings were reported by Chairman G. R. Bott, of the Ball and Roller-Bearings Division, to be proposed to bring them uptodate and simplify them. The proposal as printed last month was passed by the Committee.

Spark-Plug Length Is an Issue

Everything was moving smoothly, on the surface at least, when W. R. Strickland presented for the Electrical Equipment Division a report that emanated from a subdivision of which he is chairman. This had to do with a proposed return to the former standard of 9/16 in. for length of the base of the metric spark-plug for automobiles, and Mr. Strickland recounted how the recommendation had been referred back once before because of an objection and a thorough canvass had shown a large majority of both the Electrical Equipment and the Gasoline Engine Divisions to be in favor of the proposal, although a minority in the sub-division were opposed.

When approval of the proposal was moved and seconded, the minority became vocal in the person of O. C. Rohde, of the Champion Spark Plug Co. The threaded portion of the base, ½ in. long, was as far as the standard should go, said he; anything beyond that would limit design and is therefore contrary to the first principles of S.A.E. standardization. The length of the shell should be left to the individual designer.

Armed with the slides showing a number of engine designs which required plugs either longer or shorter than the proposed standard, he said that these represented 1,623,000 registered cars. The 9/16-in. dimension would interfere with the progress of design of the spark-plugs made by his company.

Spark-Plug Standard Is Adopted

Arguments in favor of the proposed dimensions were offered by A. Ludlow Clayden, who said it would not interfere with the production of special plugs as long as that might be necessary; by C. W. Spicer, who thought that three-quarters of the designs shown by Mr. Rohde could be adapted to the 9/16-in. length by some minor change such as in the depth of counterbore; and by Mr. Strickland, who objected to the idea of a multiplicity of skirt lengths.

Votes were taken on Mr. Rohde's motion to omit the total skirt length from the standard, which was lost by 12 to 5; on a motion to table, which was lost by 15 to 3, and then on accepting the proposal as presented, which was carried orally without question.

Serenity returned with the presentation of the report of the Lighting Division by its Chairman, C. A. Michel. All of its proposals were accepted, as presented on pp. 11 and 12 of the report printed last month. A question was raised as to why the signal lamps were not called stop lights and was answered by Mr. Michel by saying that the latter name is only a hangover from the time when all such lights bore the letters "STOP."

Reports of the Motor-Truck Division, given by Chairman Arthur W. Herrington, and that of the Non-Ferrous Metals Division, presented by R. M. Curts, were accepted without question except as to the numbering of the nonferrous metals. Mr. Spicer said that they, like the various steels, ought to be given significant numbers. He was informed by Mr. Underwood that such a proposal would be worked upon by the Division, but that waiting for such numbers to be developed would delay the adoption of the standard.

Oil Standards Are Improved

Lubricant standards came in for a little clarifying and extending as the result of a meeting of the Lubricants Division held in Detroit recently and reported by George A. Round. Gaps between the viscosity numbers are closed; the No. 60 classification is given definite limits of 105 to 125 sec.; and a new No. 70 is added, having viscosity limits of 125 to 150 sec. The last actions were recommended because of oils on the market that are materially heavier than the lower No. 60 limit. An explanatory note is added also to call attention to the fact that the S.A.E. numbers refer to viscosity only, not to other factors of quality or character.

Discussion brought out the desirability of immediate clarifying of the oil specifications because of a publicity campaign that is planned to carry these standards to the people who should benefit by them.

The tentative report of the Screw-Threads Division, in regard to round unslotted-head bolts, was withdrawn because of a patent question that has not been definitely settled.

Authorization was voted to make changes in tire and rim standards such as might be needed to make them agree with the standards that have been established by the Tire and Rim Association, subject to vote of the Tire and Rim Division.

Veterans of Standardization Speak

Routine business being disposed of, Chairman Boor called upon several veteran members of the Standards Committee to speak in honor of the 25th Anniversary. The first man called upon was Mr. Spicer, possibly the old-

est member of the Committee in terms of continuous service, who spoke briefly of the great savings resulting from S.A.E. Standards and of his enjoyment of his work upon them.

Mr. Clayden, a former chairman of the Standards Committee, credited the vision of Henry Souther, the founder of the work of the Committee, with organizing what has proved to be the mainstay of the Society and of the American automobile industry.

A. L. Riker was called upon as the first President of the Society and said that the desire for quick action in establishing standards for a new industry was one of the chief reasons why the infant S.A.E. did not merge with an older engineering society which had been unable to arrive at needed standards after years of work. But Mr. Riker's outlook was by no means confined to the past, and he said that research will replace standardization as the leading activity of the Society.

Former President H. W. Alden lauded the spirit of cooperation of the Society, recalling that he used to go to the meetings of the Mechanical Branch of the Association of Licensed Automobile Manufacturers with written instructions to "get all the information you can and give none."

Mr. Bott spoke briefly of the history of ball-bearing standardization, stressing its now international character; and R. S. Burnett called attention to exhibits of standards history in the room, including the first report on spark-plugs, made in 1908. Before adjournment, a vote was passed requesting the chairman to send a message to Mr. Clarkson, regretting his absence.

Those present at this meeting included the following:

Standards Committee Members:

A. Boor, *Chairman*

C. E. Bonnett	B. J. Lemon
G. R. Bott	C. A. Michel
R. S. Burnett	L. Ochtman, Jr.
R. E. Carlson	O. A. Parker
L. A. Chaminade	H. N. Parsons
R. M. Curts	G. A. Round
W. N. Davis	H. J. Saladin
J. F. Hardecker	K. D. Smith
A. W. Herrington	C. W. Spicer
M. C. Horine	J. G. Swain
H. S. Jandus	R. R. Teetor
W. C. Keys	A. J. Underwood

S.A E. Members and Guests:

H. W. Alden	A. L. Riker
J. A. Anglada	O. C. Rohde
A. L. Clayden	H. F. Schippel
F. H. Hazard	E. M. Schultheis
J. A. Harvey	W. R. Strickland
P. M. Heldt	J. C. Tuttle
E. S. MacPherson	L. R. Wilkinson
J. W. Oehrli	A. J. Wilson
J. E. Partenheimer	A. C. Woodbury
H. W. Perry	D. C. Young

Constitutional Amendments Approved

To Be Submitted to Mail Vote—Standards Committee Report also Approved at Business Session

MONDAY evening's business session, immediately preceding the Transmission Session, was disposed of in short order, with President Warner presiding. Mr. Warner first announced that the Army Motor Transport Unit, that had arrived earlier in the evening with its motorized equipment from Camp Holabird, Md., under the command of Major Lawes, was encamped in Dry Hollow near the hotel and that, through the courtesy of the Major, a motorcoach shuttle service would be run at half-hour intervals from the hotel to the encampment beginning on Tuesday and on the succeeding days of the meeting for the convenience of the members and guests of the Society.

The Chairman next read a telegram from Secretary Clarkson, at Briarcliff Manor, N. Y., in response to the message of greeting and regret for his absence, which, on a motion by A. Ludlow Clayden, at the Standards Session on Sunday afternoon, the members voted to send to him. Mr. Clarkson's message read:

"Greatly appreciate your message. Regret my inability to be with you. Best wishes for a fine meeting."

Annual Dues and Student Enrollment

Proposed amendments to the Constitution and By-Laws that have been the subject of discussion for several years were read in part by President Warner. These were printed in full in the January, 1930, issue of THE JOURNAL and were distributed at the session in the form of reprints. The first is an amendment to Section C 21 of the Constitution and makes a change in membership annual dues, representing an increase of $5 in all grades with the exception of Junior Member. In conjunction with this change in dues, other changes provide that the Society shall pay to the Sections $5 for each Member, Associate or Affiliate Member Representative residing in the geographical territory of a Section, which amount shall be considered as Section dues collected for the Section by the Society, and all such members in good standing shall be members of a Section. Each Affiliate Member Representative will receive the publications of the Society on the same basis as individual members and his name will be included in the alphabetical, geographical and company lists of the S.A.E. Roster. The amendment makes corresponding changes in the Section Constitution and By-Laws.

Another proposed amendment is to Section C 51 of the Constitution of the Society relating to student enrollment. It provides for the individual or group enrollment of persons under 30 years of age who are bona fide students of a recognized institution of engineering or pursuing a course of study in automotive engineering. The change involved is the substitution of the term "automotive" for "automobile" engineering so as to include aeronautic engineering.

After reading the proposed amendments, President Warner recalled the provisions of the Constitution relative to amendments, saying that after an amendment has been proposed at an Annual or Semi-Annual Meeting and discussed at the meeting, it shall be mailed by the Secretary to each voting member at least 60 days prior to the next Annual or Semi-Annual Meeting and that at the Annual or Semi-Annual Meeting the proposed amendment shall be presented for discussion and final amendment. Subsequently thereto, it shall be submitted by letter-ballot to all members entitled to vote, provided that 20 votes are cast at this meeting in favor of such submission.

PRESIDENT EDWARD P. WARNER

To Whose Energetic and Competent Work in Advance of and Participation in This Year's Meetings the Society Owes a Debt for Their Success

Prof. H. M. Jacklin then moved that

CHAIRMAN AND DISCUSSERS IN STANDARDS COMMITTEE SESSION

(1) Arthur Boor, Chairman of the Standards Committee. (2) George A. Round Reported for the Lubricants Division. (3) George R. Bott Reported for the Ball and Roller-Bearings Division. (4) C. A. Michel Made the Report of the Lighting Division. (5) B. J. Lemon Discussed the Tire and Rim Report. (6) A. W. Herrington Reported for the Motorcoach and Motor-Truck Division. (7) Otto C. Rohde Criticized the Report of the Electrical Equipment Division on Spark-Plugs. (8) Walter C. Keys Was Among Those Who Defended the Proposal To Return to $\frac{9}{16}$ In. Length of Base for Metric Plugs

the amendments be submitted by letter-ballot for final vote and the motion was seconded. President Warner called for an affirmative vote by the raising of hands. The count showed 38 votes in the affirmative and accordingly it was directed that the amendments be submitted to mail vote of all voting members of the Society.

Standards Reports Adopted

A. Boor, Chairman of the Standards Committee, reported that the Committee, at its meeting the preceding afternoon, received the reports of the various Divisions and acted upon them. The presentation of and discussion on the Division reports at the Standards Session are reported under the news account of the Standards Session in this issue of THE JOURNAL.

Mr. Boor stated that all of the reports went through as submitted with a few minor changes and very little discussion. The changes included some minor additions to the report of the Aircraft-Engine Division and there was considerable discussion of the report of the Electrical Equipment Division relative to metric spark-plugs but the report was approved.

The next procedure necessary to make the standards and recommended practices effective was for the business meeting to approve them after they had been submitted to the Council. The Standards Committee had obtained the approval of the Council at the Council meeting following the Standards Session and Mr. Boor therefore moved that the report of the Standards Committee be approved. The motion was seconded, there was no discussion, and the motion was carried by a unanimous *viva voce* vote.

President Warner thereupon adjourned the business meeting and relinquished the chair to T. J. Litle, Jr., to preside over the Transmission Session, which is herein reported separately.

Engines Lead the Way

Taub Is Headliner at First Technical Session—Dowdell Reads Paper on Bearing Bronzes

CHAIRMAN H. T. Woolson, Chrysler chief engineer, well stated the suitability of engines as the subject of the opening technical session of the Semi-Annual Meeting. The prime mover is the most important subject of all to consider, and it offers an unlimited field for research and design. Its study involves chemical ractions, heat effects, transmission forces, electrical devices and the problem of the same surface first exposed to excessive temperature and then requiring lubrication. He said that progress may seem slow, but that it is necessary to go back not more than one-third of the length of the Society's life for comparison to see that real progress is being made. Differences in engine design, Mr. Woolson sees not as a difference in what engineers think desirable but as lack of agreement as to which of the desirable features shall be sacrificed to obtain others that seem more desirable.

One unusual circumstance is that two papers were read at the same session by one man, Alex Taub, experimental engineer of the Chevrolet Motor Co. The first paper, on Powerplant Economics, is an argument for larger piston-displacement rather than higher speed or other means for making a moderate increase in power. In discussing the paper which is given in full elsewhere in this issue of THE JOURNAL, the author made it clear that he was pleading for something in the order of changing a 230-cu. in. engine to 260-cu. in. for the 10 or 15 per cent added power that is needed in a new model rather than to change the speed from 3000 to 4000 r.p.m.

A. L. Clayden spoke of H. M. Crane and F. W. Lanchester as chief advocates on opposite sides of the Atlantic, of large-bore engines. The vicious European taxation systems he blamed for small bores and long strokes. The valve troubles and troubles from excessive reciprocating weights that were common 15 years ago were solved by advances in metallurgy. Mr. Clayden doubts if the same process can be repeated now for a great further advance in speed or compression.

Higher Speeds Are Defended

Generally agreeing with the statements in Mr. Taub's paper, Robert N. Janeway spoke of a new converter development in mixture-producing devices which converts the liquid fuel into dry gas as it leaves the carbureter, with the aid of concentrated heating and absorption, so that distribution is independent of manifold velocity.

Interpretation of the rule of result per dollar must, according to H. E. Maynard, assistant chief engineer of the Chrysler Corp., be finally decided by the user, and it may be that the order of desirability of various features differs between high-priced and low-priced cars. Improvements in bearings, lubricating systems, valve materials, valve-seat hardness and other things have made possible for well-designed engines speeds of 3000 r.p.m. at full load for long periods.

Lighter engines make possible lighter units throughout the car, Mr. Maynard said. Engineers have unconsciously adopted the principle of making a pound of material work harder, and production men are making it possible to sell that pound cheaper. In the lower-price classes of Chrysler products, smaller engines have been found more economical, and four-speed transmissions are being adopted to give increased life through slower speed without sacrificing performance.

Mr. Taub argued that the adoption of the four-speed transmission is evidence of recognition of the undesirability of excessive engine speeds. The owner wants durability, and that cannot be had with oil temperatures of 275 to 340 deg. fahr. He also went after the "front-office engineers" who say that an eight-cylinder engine must be made to go in a certain space, and cause designs in which water-jackets are omitted between the valves.

Floyd F. Kishline, Graham's assistant chief engineer, considers enlarging the engine to be the line of least resistance but getting more torque, rather than power, out of each cubic inch to be the line of real progress. A. G. Herreshoff, director of truck engineering of the Chrysler Corp.; W. R. Strickland, Cadillac assistant chief engineer; and the chairman also contributed to the discussion.

Research in Bronzes

Dr. R. L. Dowdell, senior metallurgist at the Bureau of Standards, presented the paper which he had written, jointly with E. M. Staples and C. E. Eggenschwiler, on Bearing Bronzes with Additions of Zinc, Phosphorus, Nickel and Antimony. The two other authors are employed by the Bunting Brass & Bronze Co. and work with the Bureau on the research-associate basis.

The object of the research was to find the effect of various additions, which might represent incidental alloys resulting from including "secondary metals," instead of only virgin metals, in the foundry mixture. The procedure was to make up groups of samples, each headed by a commercially pure alloy and including several alloys in which were definite inclusions of the various "impurities." Samples of all these alloys were made for tests of wear resistance, resistance to repeated pounding, resistance to single impact, and hardness.

Results of the various tests on the base bronze were shown by five ternary models in plaster, in which the position of an ordinate on the base triangle indicates the proportions of the three elements in the alloy and the height of the ordinate on a certain model represents the results of tests on the alloy for the

NOMINEES FOR S.A.E. OFFICES IN 1931

(1) Vincent Bendix, for President. (2) C. W. Spicer, for Treasurer. For Vice-Presidents: (3) Dr. George W. Lewis (Aircraft), (4) Arthur Nutt (Aircraft Engine), (5) W. F. Joachim (Diesel Engine), (6) J. A. C. Warner (Passenger-Car), (7) C. B. Parsons (Passenger-Car Body), (8) A. K. Brumbaugh (Production), (9) F. K. Glynn (Transportation). For Councilors, To Serve for Two Years: (10) F. S. Duesenberg, (11) Norman G. Shidle and (12) C. R. Tilston

CHAIRMAN AND DISCUSSERS AT THE ENGINE SESSION

A. G. Herreshoff and H. C. Maynard (Upper Left and Right Respectively) Were Discussers of Mr. Taub's Paper on Powerplant Economics. H. T. Woolson (Center), Chairman of the Session. O. C. Berry (Lower Left), One of the Discussers of Mr. Taub's Paper on Combustion-Chambers. Dr. Carl Claus (Lower Right) Took Part in the Discussion of the Paper on Bearing Bronzes, Read by Dr. R. L. Dowdell

quality represented in that model. Cuts of these models are shown in the paper, which will be printed in an early issue of the S.A.E. JOURNAL.

Graphic comparison between properties of each base alloy and its derivatives is made by groups of narrow rectangles, representing ordinates for the various properties of the different alloys. Some of the tests were made both at room temperature and at elevated temperatures. Micrographs also are shown, to indicate the structure.

Conclusions Are Tabulated in the Paper

Results of the different additions on the properties were summarized in tabular form in the paper. The addition of 4 per cent of zinc increased the resistance to pounding at room temperature; the addition of 0.05 per cent of nickel had the same effect but decreased the resistance to pounding at 600 deg. fahr. and increased the wear resistance in most cases; the addition of 2 per cent of nickel made a marked increase in resistance to pounding at 70 deg., increased the impact resistance with low lead and decreased it with high lead, and decreased the wear resistance; and the addition of 1 per cent of antimony increased the frictional force at 70 deg., the resistance to pounding at all three test temperatures, and the wear resistance in most cases, but it decreased the resistance to impact by about 30 per cent. Other properties, including hardness, were not much affected by the additions in any case.

Instead of reading the paper in full, Dr. Dowdell outlined the methods of testing, giving photographs of some of the apparatus, charts of the results and a table of conclusions, thus making the presentation briefer and more interesting than the full paper.

Maintaining Oil Film Is Essential

Dr. Carl Claus, experimental engineer of the Bound Brook Oil-Less Bearing Co., spoke of the necessity of maintaining an oil film and of a method of producing bearings by mixing mechanically a finely pulverized metal with a small percentage of graphite which is then compressed and heat-treated at a temperature below the melting point of copper. The resulting porosity helps to maintain an oil film.

Such bearings are useful for light work, Dr. Dowdell said; but their suitability for heavy pressures is doubtful, and their economics is more of a question. He reported a French white bearing-metal into which graphite is introduced under pressure. Dr. Claus admitted that the synthetic bearings required more expensive material than do castings, but the quantity of material and the manufacturing cost are reduced by compressing the bushings to very close limits on automatic machines.

Detonation Theories Coordinated

H. R. Ricardo published a paper in 1919 which is credited by Mr. Taub with being responsible for making the world combustion-chamber conscious and inspiring other engineers to work on the detonation problem. Outstanding among these other workers are R. N. Janeway and W. A. Whatmough. The third paper presented in the engine session was a comprehensive attempt on the part of Alex Taub to correlate the work of these three men, and to find from their works, if possible, the common ground which engineers who have not studied the question so deeply can accept as known. Mr. Taub finds that the principle of cooling the last gas to burn, as propounded by Mr. Janeway, is followed by both of the other investigators, and that Mr. Ricardo has ceased to insist on turbulence as an important preventive of detonation and has also enunciated the importance of cooling the last gas to burn.

Because his paper was too long and involved to be read and assimilated in such a session, Mr. Taub presented it in abstract form, with the most important conclusions and illustrations.

The paper brings out many apparent differences in the works of the three authors under discussion and shows that many of them can be reduced to one set of facts.

Ricardo Pioneered the Study

High praise is given to Mr. Ricardo by Mr. Taub who says that, if the early Ricardo turbulent head had not made engines rough, Mr. Janeway and others would not have had the incentive to strive for smoothness. Mr. Whatmough is credited with being a physical chemist of real analytical ability, who has influenced English practice for the better, but the greater praise is given to

Mr. Ricardo and Mr. Janeway; the former as the pioneer and a thorough technician who goes further than any other man or organization to prove or disprove his theories, and the latter for evaluating the fundamentals and developing a yard-stick by which combustion roughness can be evaluated and compared. Mr. Taub says that this work will be accepted as one of the finest contributions that has been made to the thermodynamics of the internal-combustion engine in recent years of investigation.

Discussion on this paper was not so lively and acrimonious as it might have been if Mr. Whatmough had been on this side of the Atlantic, as he was a year ago. Chairman Woolson said that a preprint of the paper had already started the pencil of one engineer to sketching his ideas for an efficient combustion-chamber.

Carbureter Problems Are Related

Attention was called by O. C. Berry, director of engineering of the Marvel Carburetor Co., to the interest of the carbureter engineer in combustion-chamber design. When a vacuum of 19 in. of mercury is induced in the inlet manifold, there is a rush of burned gases from the cylinder into the inlet manifold. Only during the last part of the inlet stroke, therefore, is a good mixture drawn into the cylinder; and, if the design is not such that this good charge is concentrated at the spark-plug, bucking will result during idling.

Mr. Kishline reported experience that indicated little effect on idle running from changes in spark-plug location but bad effects from opening the inlet valve of a cylinder before the exhaust valve thereof was closed, because of gases drawn back from the exhaust manifold.

A. W. Pope, of the Waukesha Motor Co., confirmed some of Mr. Taub's statements and said that materials, rather than induction systems, are limiting the speed of heavy-duty engines, in which life and durability are foremost requirements.

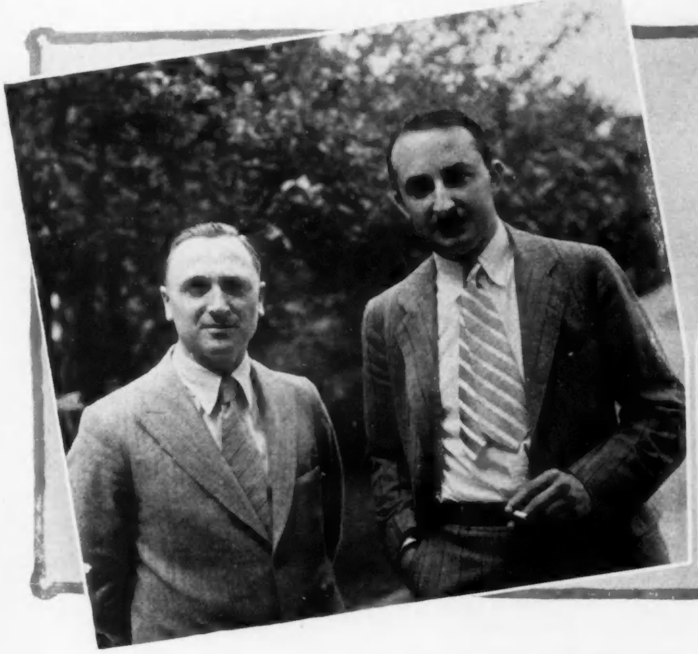
In closing, Mr. Taub cited difficulties with the Chevrolet cylinder-head when adapting it for an export model. Changing conditions to give a better mixture around the spark-plug was found to be the remedy.

Diesels Draw Full House

Hill and Cummins Describe Their Engines and Tell Experiences—Fuel Possibilities Stated

ONE of the high-lights of the meeting was the Diesel-Engine Session of Tuesday morning. An attendance of well over 100 crowded into the smaller or north hall to hear and discuss papers by H. D. Hill and C. L. Cummins, each at the head of a Diesel-engine-manufacturing concern bearing his own name. Chairman A. J. Poole, of the Robert Bosch Magneto Co., announced that two Diesel-engined vehicles had been driven from the Metropolitan area for the meeting: a motorcoach of the Public Service Corp. of New Jersey, which had made the trip from Newark on fuel at the rate of 7 miles per gal., and a 5-ton truck that had averaged 15 miles per gal. for the trip from Long Island City. Those in charge invited inspection and offered demonstrations.

Precombustion-chamber burning, and particularly the Hill antechamber system, were described in Mr. Hill's paper, and their advantages were stated. Fuel systems that are satisfactory in cylinders of 8-in. bore or larger cannot be simply reduced in size for high-speed engines of 5-in. bore or smaller. A higher degree of accuracy, for timing and metering, and preventing the spray from striking the cylinder-walls are



TWO HEADS THAT DO NOT KNOCK

Alex Taub (Left) Read Two Papers in the Engine Session. Robert N. Janeway (Right) Discussed the First of Mr. Taub's Papers, and Mr. Janeway's Writings on Combustion-Chambers Were Among Those Reviewed in Mr. Taub's Last Paper

MILWAUKEE MEMBERS INSPECT DIESEL MOTORCOACH

Fred M. Young (Left), of the Young Radiator Co., Discussed Radiators for Diesel Engines. A. W. Pope, Jr. (Right), of the Waukesha Motor Co., Spoke on Combustion-Chamber Design in the Engine Session

among the difficulties encountered in such an adaptation.

Reducing Sensitiveness of Diesel Engines

Direct fuel injection requires very small orifices and hydraulic pressures which Mr. Hill said might need to be as much as 8000 lb. per sq. in. Larger orifices and lower fuel pressures can be used with pre-combustion chambers, the injection pressure in the Hill engine being only 1500 lb. One distinctive feature of Mr. Hill's design which he described is that connection between the antechamber and the cylinder is through one relatively large opening, instead of through several smaller ones. This allows flame to pass into the cylinder space without being quenched. For another thing, fuel does not enter the upper part of the chamber, so that air for combustion is fed to the cylinder from this source and causes turbulence in the cylinder.

Other features of his engines which Mr. Hill described and illustrated included the injection nozzle, which includes a spring-loaded valve, and the fuel pump. The fuel consumption of 5 x 7-in. and 6 x 10-in. models is said to be from 0.42 to 0.46 lb. per hp-hr., according to the heat value of the fuel.

How fixed is the status of gasoline engines for automotive work was illustrated by Mr. Cummins in his paper by the difficulties that he had in securing a license and insurance for his first Diesel-engined cars. After issuing a policy, one of the leading insurance companies cancelled it because they were unfamiliar with the hazards that the Diesel engine might develop in a car. A policy was finally issued on condition that no one but Mr. Cummins and his brother should drive the car.

Establishing an Official Speed Record

The car used in Mr. Cummins' trip to New York, which has been reported in the S.A.E. JOURNAL already, was a seven-passenger limousine-sedan, which weighed about 6000 lb. and was driven 6000 miles with a fuel consumption ranging from 25 to 35 miles per gallon, according to the speed.

The car used at Daytona Beach was a roadster of the same make, weighing 4500 lb. With the same 2½:1 axle ratio as on the first car, the roadster travelled the 1060 miles from Columbus, Ind., to Daytona Beach on 39 gal. of fuel. An official record of slightly over 80 m.p.h. and a single run at about 88 m.p.h. under more favorable conditions were made, using a gear ratio of 2:1, which was retained when the car was driven back to Indiana.

Mr. Cummins described the construction of his engines, which were described in a paper by him printed in the S.A.E. JOURNAL for October, 1927, p. 388. The chief change made since then has been the addition of a small bottle in the top of the piston, having an outlet opposite the injection nozzle. This discharges

PROMINENT AT THE DIESEL-ENGINE SESSION

A. L. Poole, Chairman, Invited Inspection of Diesel-Engined Truck from Long Island City

C. L. Cummins Described His Engine Design and Record-Making Runs at Daytona Beach, Fla.

Charles O. Guernsey Was One of the Discussers of the Papers by Mr. Hill and Mr. Cummins

A. A. Lyman Brought a Diesel-Electric Motorcoach from Newark and Described Its Performance

A. Ludlow Clayden Discussed Fuels in the Diesel-Engine Session and Many Subjects in Other Sessions

A. L. Beall Had Something To Say about Fuels and Oils in This and Some of the Other Sessions

NEW TYPES OF MOTOR-VEHICLE SEEN AT THE SUMMER MEETING

(1) A New Small Partly Streamlined Motorcoach. (2) Diesel-Electric Motorcoach of the Public Service Co. of New Jersey, Driven to the Meeting from Newark. (3) Diesel-Engined Motor-Truck of the Robert Bosch Magneto Co., Driven to the Meeting from Long Island City, N. Y. (4) Cummins Diesel-Engined Packard Car Driven to the Meeting by C. L. Cummins. (5) Benz Diesel Engine Used in the Public Service Co. Motorcoach. (6) M.A.N. Diesel Engine Used in the Bosch Company's Truck

into the overrich region around the injection nozzle air which is forced into the bottle during the compression stroke and prevents carbonization of the nozzle.

Mr. Cummins announced that his organization is now designing a Diesel engine comparable with gasoline engines used in motor-trucks and motorcoaches, with which he expects to make a rapid entry into the automotive field.

Motion pictures taken on Mr. Cummins' Florida trip were shown after the presentation of his paper. These showed Mr. Cummins and his Diesel-engined Packard roadster passing through various towns along the route, as well as on the beach at Daytona. Preparations for the race were shown, and the car was seen running in each direction along the water's edge. The Diesel-engine runs were made while Kaye Don was working at the beach, and some of the films showed him with the Silver Bullet.

Diesel-Engine Fuel

Discussion of the two papers came after both had been read. A. L. Clayden, of the Sun Oil Co., opened with a warning against the tendency to build a fools' paradise based on cheap Diesel fuel. He said that there is potentially less Diesel fuel than gasoline in a given amount of crude oil. Diesel engines have plenty of advantages, he said, without using incorrect arguments about low-cost fuel. Questions in regard to standardization of Diesel fuel which he brought up include the several viscosities required and whether lubricating qualities, which would add to the cost, are needed.

Mr. Clayden made the interesting statement that he and his associates do not and will not make aviation gasoline, one reason being that they believe that the day of the use of gasoline as an aviation fuel is rapidly coming to an end.

He was answered by both authors. Mr. Hill said that only about 10 per cent of the present crude is gasoline without cracking, which adds to the cost.

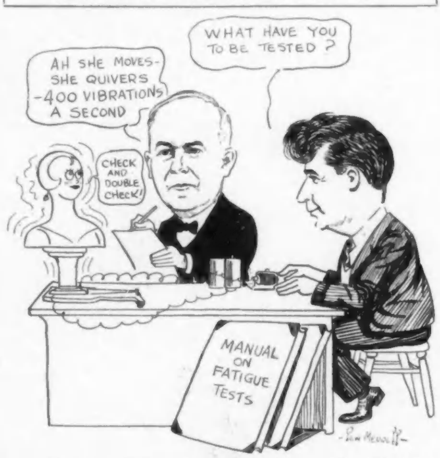
Mr. Cummins said that he expects the cost of fuel oil to go up, but crude oil from some fields is suitable for use as Diesel fuel after no treatment except centrifuging. This will make it possible to buy fuel directly from the producer if the refiner should make the cost unduly high. A large part of the production is by independents who have no refinery connections and would welcome an independent market.

Past-President T. J. Litle, Jr., and H. E. McCray, chief engineer of the Waterloo Gasoline Engine Co., started a game of 20 questions that kept Mr. Cummins somewhat busy for a while. He stated his belief that the day of the automotive Diesel is at hand, and that it will weigh about 10 per cent more and cost about 15 per cent more than a heavy-duty gasoline engine of the same power. Acceleration, Mr. Cummins says, is better with his Diesel car than with lively gasoline cars.

Throttling, Mr. Cummins believes, can be made quite satisfactory in a Diesel engine designed for automotive service. The engine he has used, having its flywheel reduced to go inside the Packard flywheel housing, will throttle to about 350 r.p.m. satisfactorily, but the performance is somewhat rough if the speed is too low.

Fred M. Young, of the Young Radiator Co., said that his experience with radiators for Diesel engines was that they required two or three times the capacity that is needed for a gasoline engine of the same power, and asked Mr. Cummins to explain his observations of easy cooling. Mr. Cummins thought something must be wrong with a Diesel that was so hard to cool, but Mr. Young maintained that 95 per cent of them fail to cool easily.

The Professors Have Their Inning



The Professors, Moss and "Gallant" Fox, have a serious job. Don't smile at the Wanamaker dummy, unsuspecting reader—it may be your turn next. Future generations, that are destined because of you engineers to be 100 percent automobile riders, must be made comfortable even if the present generation does suffer the discomfort of a wobble-meter test.

Mr. Cummins said that he has operated Diesel engines on gasoline, but this is not a suitable fuel and requires an addition of lubricating oil. Mr. Hill told of some auxiliary pumping stations on pipe lines that are driven by his engine running without difficulty on the Illinois and Oklahoma crude oils. The main engines in these plants are of DeLavergne make, and the fuel carries so much silt that it had filled a gate-valve when it was dismantled recently. Mr. Hill said that direct fuel-injection systems usually start easier than precombustion engines, and engines of the latter type having small chambers usually start harder than those with large chambers. A cranking speed of about 175 or 200 r.p.m. is required for starting.

Charles O. Guernsey, chief engineer of the automotive car division of the J. T. Brill Co., also launched a series of questions, directed at the readers of both papers. Mr. Hill said that the true Diesel cycle will not operate fast enough for small high-speed engines, which are generally made of the combination cycle in which the explosion pressure is higher than the compression pressure. Mr. Cummins said, in regard to the field of Diesel engines, that they are being used mostly for service where large amounts of fuel are used. Small Diesel engines are used in lighting service in small towns after the load is so light that the large main engine does not function properly, and in still smaller towns where they are sufficient for the entire electrical requirements. The Government is now buying only Diesel engines for lighthouse service. Numerous installations have been made in buildings for emergency use and also for producing the electricity consumed in a building at much lower cost than it can be bought. In one building which contains a garage, the cooling water from the engine is used for heating the building and, after passing through the radiators, to wash cars. The oil drained from crankcases is used for fuel.

A. A. Lyman, of the Public Service Coordinated Transport of Newark, N. J., offered remarks based on operating experience with the Diesel-engined motorcoach which was shown at French Lick. In actual service during the winter, this 1800-lb. electric-drive vehicle consumed fuel at a rate of 5 to 5½ miles per gal. An identical vehicle equipped with a gasoline engine operating on the same route averaged about 3.3 miles per gal., showing a saving of two-thirds or 2 cents per mile in fuel cost for the Diesel. This will amount to about $1,000 on a vehicle running 50,000 miles per year. Exhaust-gas analyses made by both Mr. Lyman and an engineer of the Holland Tunnel showed a maximum of 0.7 per cent carbon monoxide under any condition of speed or load. This motorcoach has been operated by a large number of drivers. Many of them have received no instruction other than to step on the heater switch which connects the glow plugs before they step on the starter switch.

Daily Meeting News

DAY-TO-DAY news of happenings at the meeting and programs of each day's technical sessions and social events were provided at the breakfast table as usual at Summer Meetings through the medium of the *Daily S.A.E.*, which was printed and distributed this year through the courtesy of the Houdaille-Hershey Corp., of Chicago. The competent staff of editors supplied by the corporation was comprised of Ray W. Donahue, editor-in-chief, Fred W. Miller, Art B. Heiberg and Stanley M. Black. These were

assisted by the following associate editors: D. I. Cook, George V. Foy, Louis Schwitzer and B. M. Short.

As a sporting proposition, this year's *Daily S.A.E.* blossomed out as a pink sheet and was replete with cartoons and halftone portraits in addition to newsy personal items of members in attendance. Several of the cartoons are reproduced in connection with the full report of the meeting in this issue of the S.A.E. JOURNAL.

News of daily events in the automotive industry in general was brought to the meeting in copies of *The Automotive Daily News*, printed in New York City and flown to the French Lick Springs Hotel in the Prest-O-Lite Co.'s airplane, which made its landings on the lower golf course. Copies of the *News* were distributed gratis.

Complimentary copies of the *Chicago Daily Tribune* were also available each day in the hotel; so, although French Lick is far off the beaten track, no member attending the meeting for the week could feel out of touch with the happenings in the automotive industry and in affairs in general throughout the Country and the world.

Aircraft-Engine Session

Vapor Lock and Engine Installation Engage the Only Aeronautic Session of the Meeting

WEDNESDAY morning, the session on Aircraft and Aircraft Engines occupied the main meeting room. George W. Lewis, director of aeronautical research for the National Advisory Committee for Aeronautics, was in the Chair, and President Warner offered a resolution concerning the death of Mr. Woolson, which was passed, as follows:

Tribute to L. M. Woolson

Resolved, That the Society of Automotive Engineers, assembled for its aeronautic session, sorrowfully takes notice of the tragic accident in which our admired and beloved fellow-member, Lionel M. Woolson, lost his life.

That the Society wishes to express its appreciation of the great loss that has been suffered by its own councils, in which Captain Woolson was for a number of years an active and inspiring member, and by engineering science, in which he had been for many years a leader. That our profound and sincere sympathy is with Captain Woolson's family. Our sense of loss in the death of our friend and co-worker is only less acute than theirs. That this resolution shall be printed in the Journal of the Society, and spread upon its records, and that copies, suitably prepared, shall be forwarded to Mrs. Woolson.

The first paper to be presented was that on Vapor Lock, by Dr. O. C. Bridgeman and H. S. White, of the Bureau of Standards. In presenting the paper, Dr. Bridgeman said that the problem of vapor lock cannot be confined either to the petroleum industry or to the automotive industry; it must be solved by cooperation between the two. A fuel line can be designed that will give vapor lock with any fuel, and a refiner can make a fuel which will give trouble with any line.

Vapor Pressure Controls Locking

When fuel starts to boil, vapor lock may result. The first phase of the investigation was a study of the conditions under which fuel boils, and vapor pressure is the factor which determines when gasoline starts to boil. The general conclusion was that the 10-per cent point of the standard A.S.T.M. distillation test of most gasolines is the temperature which determines whether vapor lock is likely to occur. Satisfactory aviation fuels contain little propane. Vapor lock may occur at temperatures lower than the 10-per cent point with automobile fuels, which are more likely than aeronautic fuels to contain propane.

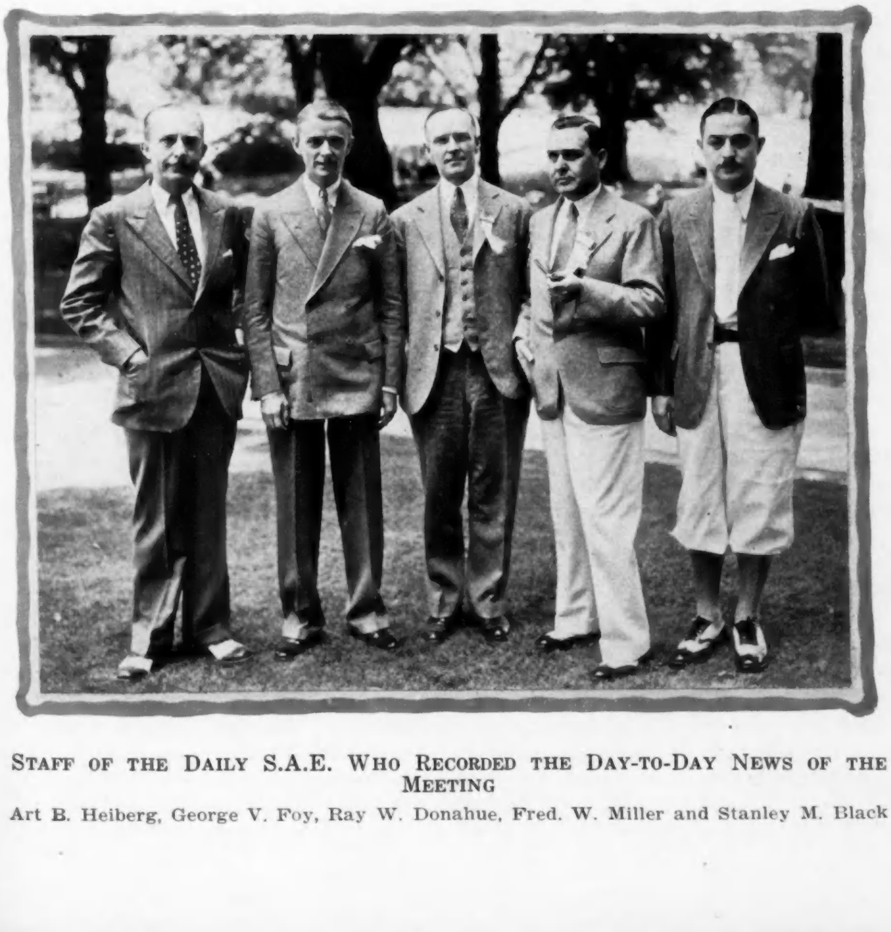
A series of experiments was made with various gasolines flowing by gravity through different arrangements of piping under various conditions. The temperature is gradually raised during the test and the flow is measured. The temperature at which the flow begins to change is called the vapor-locking temperature. Curves shown indicate that a small change in temperature at the critical point may reduce the flow by 40 or 50 per cent and cause the engine to stop.

Experiments were made with gravity flow from an orifice, with and without a float chamber inserted in the line, some of them with heat applied to the line between the float chamber and the orifice; also flow under gravity to carbureters of standard makes and flow to a gear fuel-pump with a suction lift.

One of the interesting observations was that a sudden increase in the cross-sectional area in the feed line in the direction of flow may have a marked effect in decreasing the flow, and therefore should be avoided in actual systems. A decrease in area has no such serious effect, so long as the smaller area is large enough, but sharp bends may have considerable effect at low pressure.

In the experiments with the fuel pump, flow is entirely interrupted at temperatures considerably below the computed temperatures at which interruption was expected, the difference

(*Continued on p. 836*)



STAFF OF THE DAILY S.A.E. WHO RECORDED THE DAY-TO-DAY NEWS OF THE MEETING

Art B. Heiberg, George V. Foy, Ray W. Donahue, Fred. W. Miller and Stanley M. Black

Chronicle and Comment

In Retrospect and Prospect

ANNIVERSARY celebrations provide the occasion for reviewing the progress that has been made and of considering what should be done in the future. It is a time of stocktaking; of questioning whether our accomplishments have measured up to our expectations; and of considering the problems still confronting us and the course of action that should be adopted for meeting and solving them.

The Summer Meeting of 1930 was true to form in these respects and was an unqualified success. There were many reasons for justified self-complacency for the record of the first 25 years of existence of the Society, which have been fruitful of much good to the automotive industry and to mankind, not only in our own Country but the world over. At the same time, the technical sessions all dealt with immediate problems to which the engineers are eagerly seeking the answers, and the founders of the Society and others who stand high in its councils took the opportunity to point out the apparent desirable course of activities in the years to come.

One of the major reasons for organizing the Society has now been largely satisfied, so far as the automotive industry is concerned; standardization has been carried a long way in the motor-car field and considerable has been accomplished already toward standardization in aeronautics—sufficient, at least, that the importance of carrying on this work energetically before aircraft manufacture gets into mass production is generally realized. In the last few years standardization has entered the broader and very important fields of National and international standards.

Research, on the other hand, is growing fast in importance, pointed out one of the founders and Past-Presidents; we need to have much more fundamental knowledge if we are to carry much further the improvement in efficiency, economy, safety and comfort of motor-vehicles, aircraft and motorboats. The search for such knowledge is likely to carry us far afield from our own particular branch of engineering; important facts that may have a direct bearing on our own work are being developed in various industries.

Political economy as well as the economics of production is a field in which the Society should, perhaps, become actively engaged, suggested another Past-President. If better transportation is a boon to humanity, we need all possible facilities for it, and they must be provided by or with the aid of government. Federal and State lawmakers and authorities will welcome honest and constructive cooperation.

Engineering has many ramifications; it is broader than mechanics and electricity; it embraces economics and sanitation and conservation of life and materials. We are living in a mechanized age—the age of the engineer's greatest opportunities. The Society is still young, measured by the average age of its whole membership, and great opportunities for further important accomplishments lie in the next 25 years to which we are now looking forward hopefully.

Passing of the First President

MEMBERS who attended the Summer Meeting and noted with pleasure the active part that Andrew L. Riker, the first President of the Society, took in the 25th Anniversary celebration, were greatly shocked by news reports of his sudden death on the Sunday following the meeting. The Society really owes its existence to Mr. Riker, as is obvious from the very interesting address he made at the Anniversary Session, and throughout the first 25 years of its existence he never lost his keen interest in its work.

Not only does the Society, as an organization, deeply mourn the passing of one of its oldest members, in point of service, and most loyal friends, but all who had been personally associated with Mr. Riker in any capacity feel that they have lost a near and very dear friend who was greatly admired and respected for his humanistic qualities as well as his undoubted ability as an engineer.

It is a source of real satisfaction that the Society had the opportunity to render its honor and respect to Mr. Riker at the meeting just before he was taken from among us and to know that he enjoyed to the full his mingling again with his old and new confreres. It is not vouchsafed to many to see such full fruition of the efforts of a lifetime and to have one's accomplishments accorded such general recognition while he still lives.

S.A.E. Transactions for 1929

PRINTING and binding of the 1929 volume of S.A.E. TRANSACTIONS was being actively prosecuted at the time this issue of THE JOURNAL went to press. Copies will, it is expected, be mailed by July 1 to those members who placed orders for them at the time of paying their Society dues for the current fiscal year. A small surplus stock will be available to take care of members whose copies fail to reach them, or others who may have neglected to place an order.

This volume, which covers the entire year, will contain 91 papers and discussions presented at Society and Section meetings and articles printed in THE JOURNAL. Not all of these papers, discussions and articles are reprinted in the S.A.E. TRANSACTIONS for that year, the policy of the Publication Committee being to approve for reprinting only such material as has permanent engineering value. That the character of the papers appearing from month to month in THE JOURNAL is improving is evidenced by the fact that the Committee rejected only approximately 25 per cent of the papers, discussions and articles submitted for consideration.

S.A.E. Office to Close for Vacations

THE headquarters of the Society in New York City are to be closed from July 28 to Aug. 11 for a simultaneous vacation of the staff. This arrangement proved most satisfactory in the summers of 1928 and 1929 and is to be repeated. Several members of the office force will remain on duty to take care of any emergency or very pressing matters, but all routine work will be suspended for the two-week period.

How the S. A. E. Started and Grew

Seed Planted in 1902 and Nurtured by Far-Sighted Engineers into a Sturdy Tree Has Many Branches Grafted onto Main Stem

ORGANIZING a society of automobile engineers was suggested in an editorial published in the *Horseless Age* of June 4, 1902. Some interest was aroused, as evidenced by letters appearing in its columns during the remainder of that year, but no active steps were taken until the summer of 1903, when E. T. Birdsall wrote to prominent men in the industry, including H. W. Alden, Henry Ford, H. P. Maxim, A. L. Riker and A. H. Whiting, and advocated the formation of such a society. As outlined by Mr. Birdsall, the membership would not be confined strictly to any one group, as was that of the Association of Licensed Automobile Manufacturers, but would embrace engineers of the entire industry.

A short time after Mr. Birdsall sent out his letter, an informal meeting, which was attended by six of the men to whom he had written, was held at the rooms of the Automobile Club of America, in New York City, and the decision was reached to form a Society of Automobile Engineers. This was the first of a series of meetings at which the constitutions and by-laws of the different organizations of which the men approached by Mr. Birdsall were members were gone over. By the end of November, 1903, sufficient progress had been made to warrant the sending out of a second letter calling a meeting during the week of the 1904 New York Automobile Show to complete the formal organization of an automobile engineering society.

ANDREW L. RIKER
President, 1905 to 1907

On Jan. 18, 1904, a meeting was held at the Hotel Navarre, in New York City, at which the preliminary organization of the Society of Automobile Engineers was effected. Those present included E. T. Birdsall; H. M. Swetland, whose cooperation and interest Mr. Birdsall had enlisted; and Allan H. Whiting, who were constituted a committee to draft a constitution.

Officers elected at that time were A. L. Riker as President and E. T. Birdsall as Secretary-Treasurer. As much of the preliminary work had already been accomplished at the meetings held the previous autumn, this committee was shortly able to present a constitution that provided for the enrollment of engineers who were engaged in the designing and construction of automobiles as members of the Society, the amount of annual dues to be collected and the election of officers, including a Board of Managers.

Pursuant to the provisions of this Constitution, an organization meeting was held in January, 1905, at New York City. The officers elected at that meeting were President, A. L. Riker; First Vice-President, Henry Ford; Second Vice-President, John Wilkinson; Secretary-Treasurer, E. T. Birdsall; and Managers, H. W. Alden, L. T. Gibbs, H. P. Maxim, H. M. Swetland, H. Vanderbeek and A. H. Whiting. At that time the Society had approximately 30 members.

By the time of the first formal meeting, which was held on Jan. 15, 1906, at the New Grand Hotel, New York City, the membership had increased to over 50. A banquet, attended by 26 members and 6 guests, preceded the technical session, over which President A. L. Riker presided and at which three papers were presented. These were, Materials for Motor Cars, by Thomas J. Fay; Ball Bearings, by Henry Hess; and Some Requirements for Carbureter Design, by E. T. Birdsall.

Following the presentation and discussion of the papers, a brief business meeting was held at which the 1905 officers were reelected. The Secretary's report showed that 54 members had been elected during the previous year, 1 had died and 1 had resigned, leaving 52 names on the membership roll, 49 of whom were Members and 3 were Associates. The Treasurer reported receipts of \$390.00 and disbursements of \$46.94, leaving a balance on hand of \$343.06.

H. M. Swetland brought up the question of distributing the proceedings of the meeting to the members in printed form and made a motion to that effect which was seconded and carried. This marked the beginning of the publication work of the Society, the progress of which is given elsewhere.

At the second annual meeting, on Jan. 17, 1907, which was also held at the New Grand Hotel, New York City, technical sessions were held in the afternoon and evening, with a dinner between them which was attended by 44 members and guests. Thomas J. Fay was elected President, E. T. Birdsall was reelected Secretary-Treasurer, and the Board of Managers was also reelected. The membership by that time had reached the 100 mark. During the early part of the year Mr. Birdsall moved his business to Detroit, and the Secretary's office went with it. A campaign to secure members was started under the direction of an associate editor of *The Automobile*, C. B. Hayward. Although this resulted in

THOMAS J. FAY
President, 1908

S. A. E.

doubling the membership in one year, lack of cohesion between the Secretary's office and the other officers, augmented by the long distances between the respective offices, left the Society in a much disorganized condition.

Salaried Secretary Secured

Two decisions that exerted a great influence on the future of the Society were made at the 1908 Annual Meeting. These were to move the Secretary's office to New York City and to secure the services of a Secretary for the whole or a part of his time. Following the election of Henry Hess as the third President of the Society, Alexander Churchward was appointed Secretary by the Board of Managers on a part-time basis and the headquarters of the Society were established in his office.

During 1909 the membership of the Society was increased considerably, the total enrollment reaching 400 by the end of that year. Formal incorporation of the Society under the laws of the State of New York, increasing cost of meetings, other expenses and a desire to have the organization take full advantage of its opportunities led to a consideration of the desirability of placing the direction of the Society in the hands of a General Manager. Shortly after the election of Howard E. Coffin as President in 1910, H. M. Swetland, Chairman of the Finance Committee, met him upon two occasions and insisted that the Society employ a competent active Secretary who should devote his entire time to the service of the organization. After the second interview Mr. Coffin readily agreed to the proposed plan and suggested that Coker F. Clarkson be secured for the post. Mr. Clarkson had been connected with the Association of Licensed Automobile Manufacturers since 1905 as secretary of its Mechanical Branch, publicity manager and assistant general manager and was thoroughly familiar with the various activities of that organization, the Mechanical Branch of which had become somewhat dormant largely on account of the progressiveness of the Society. Mr. Swetland and the other Managers readily concurred in this recommendation, and Mr. Clarkson was engaged as Secretary and General Manager under the direction of the officers and Managers.

Howard E. Coffin
President, 1910

Horace M. Swetland
To Whose Active Support the Organization of the Society Was Mainly Due

The General Manager immediately assumed his obligation, collected what scattered records of the Society were then in existence, including the names of the members in good standing and otherwise, and took possession of the bank account, which then approximated $250. After acquainting himself with the facts concerning the Society and its proposed operation and looking over the possible avenues of income and expense, Mr. Clarkson communicated to President Coffin some skepticism as to the ability of the Society to pay expenses, including the salary of the Secretary. The latter's reply was typical—a personal check, with instructions to use it and call on him for more money when need arose. To obviate that necessity, Henry Hess, A. L. Riker and H. M. Swetland joined with Mr. Coffin in financing the Society and received promissory notes bearing a low rate of interest.

Henry Hess
President, 1909

Clarkson Takes Aggressive Steps

Before the end of 1910 the wisdom of securing an active Secretary was evident. Mr. Clarkson began a campaign for new members and organized the work of the Society through various committees, which has resulted in the most extensive collaboration and standardization undertaken by any engineering society of recent date. Such interest was stimulated by these activities that memberships began to pour in, and the finances, general policy and activities of the Society assumed sound and substantial proportions during this period.

At the meeting of the Council, formerly the Board of Managers, in New York City on March 15, 1910, standardization of automobile parts was decided upon and the Society took over the work begun by the Association of Licensed Automobile Manufacturers. A Standards Committee was appointed at the next Semi-Annual Meeting, with Henry Souther as Chairman. This Committee was made up of different Divisions, the personnel of which, as a rule, was equally divided between representatives of producers and consumers. The earliest work related to standardizing lock-washers, seamless tubing, frame sections, carbureter flanges and annular ball-bearings, writing iron and steel specifications and studying aluminum and copper alloys.

In 1911 the Society paid its first official visit to Europe. The first stop was

25TH ANNIVERSARY

HENRY SOUTHER
President, 1911

made in England, and then the journey was continued to France. A most cordial and very elaborate reception was accorded the members of the Society, who were admitted to the most famous factories which had never been opened to visitors before. This visit was returned two years later when a party of British engineers came to

HENRY F. DONALDSON
President, 1912

America and attended the Semi-Annual Meeting of the Society. At the conclusion of the Meeting the party went on an inspection tour of American automobile factories.

The membership had, by 1913, increased to 1683, and intensive efforts were being made toward standardizing the automotive industry. Much energy was being expended by the Society in the motorboat and stationary internal-combustion-engine fields as well as along the line of automobile mechanics.

Automobile engineering moved forward rapidly during 1916, and the Society was recognized more and more as an international factor in the efficient establishment of the automotive industry. The proposal that the activities of the Society be broadened to embrace all engineers engaged in the development of apparatus using the internal-combustion engine was favorably received by the members and by organizations in allied fields. This led to the merger of the Society of Aero-

HERBERT W. ALDEN
President, 1912 and 1923

nautic Engineers and the Society of Tractor Engineers with the Society, the taking over of engineering and standardization activities represented by the National Association of Engine and Boat Manufacturers and the National Gas Engine Association, and the change in name to the Society of Automotive Engineers.

War Work of the Society

When the United States entered the World War in 1917, the Society was put to perhaps its greatest test of usefulness. At that time the Government requested different societies to design certain units in the shortest possible time. One of the demands made upon the Society was by the War Department for a 3-ton truck having a 5-ton capacity. Specifications, prepared by the War Department and published in May, 1917, outlined certain fundamental items that were considered of vital importance for military trucks. Although these specifications called for some radical features in design which

HOWARD MARMON
President, 1913

could not be made in existing models of trucks, a number of manufacturers took steps to produce vehicles that would meet the most important, if not all, of these detailed requirements. Almost as soon as the work was started,

(*Continued on p.* 692)

HENRY M. LELAND
President, 1914

Recollection and Prophecy

TWENTY-FIVE years ago a few young engineers who were interested in developing the automobile met to consider the formation of a society that would deal directly with the engineering problems in connection with the motor-car. While perhaps all of them were members of the American Society of Mechanical Engineers, they felt that this Society did not cover specially the automobile industry from an engineering standpoint.

At a meeting held in the club rooms of the Automobile Club of America in April, 1905, at which were present not more than four or five men who were connected with the industry, the Society of Automobile Engineers, as it was then called, was formed. The Society had to make a definition of the meaning of the word "automobile," which could not be found in the dictionary. It was finally decided that the word covered any self-propelled vehicle running on the land, in or under the water or in the air.

Passing through the throes of securing sufficient members, which was a rather difficult task at that time, since the automobile industry did not support the Society as it has in later years, and having to call on some of the members to act in the capacity of officers of the Society, which they did at perhaps a sacrifice on their part, the Society increased in membership, size and importance as time went on and it was finally decided that we had reached a point where a paid secretary and manager should be appointed. We were fortunate in obtaining the services of Coker F. Clarkson for this position, and I feel that it is largely through his efforts that the Society has attained the position it now holds in the engineering world.

The organizers of the Society felt that, being connected with a new industry, they might be able to shape the progress of that new industry and that nothing better could be attempted than to ascertain if it was possible to develop a line of standards that would enable the manufacturers of motor-cars to purchase from various concerns the auxiliary parts entering into their product which would be interchangeable. I think it is a recognized fact today that the S.A.E. Standards are acknowledged all over the world.

One of the first items that the Society attempted to standardize was the spark-plug. At a meeting held at Niagara Falls, which was attended by the engineers of the various automobile companies, it was finally decided that it was not the spark-plug which should be standardized, but the spark-plug hole; therefore at that meeting the $\frac{7}{8}$-in. 18-thread spark-plug hole or spark-plug was adopted as the standard for this Country.

My personal feeling is that this and the other standards proposed and adopted by the Society have been a vital element in the success of the automobile industry in this Country. I well recollect attempting in the early days to obtain alloy steels from the large steel companies, which considered it a joke to be asked to produce steel other than what they were then producing; their comment being that the automobile was a fad and the industry would never amount to enough to warrant their manufacturing any other steels than those they were making at that time.

The picture is changed today; the automotive industry is the largest consumer of steel in the United States, and the steel companies today are endeavoring to cooperate with the automotive engineers whenever a suggestion is made for improvement in the quality of their material. The influence of the Society on other industries supplying material and parts for the manufacturer of motor-cars also has had its effect.

In the early days the question of pneumatic tires and rims was of great moment. I recollect the time when each tire maker had his own make of rim and tire size and would not guarantee his tire if used on any other make of rim than his own.

It was through the influence of the Society that a standard form of rim was adopted, on which all tire makers were willing to mount their tires and guarantee them.

I am convinced that the Society, through its activities, has made it possible for the automotive industry to produce the type of car that is being sold today at a figure that is almost unbelievable when considered in comparison with the cost of other mechanical appliances. I believe that, if it had not been for the S.A.E and its adoption of standards, it would be impossible to purchase a complete motor-car, fully equipped, for less than 40 cents per lb. Therein lies the contribution the Society has made for the benefit of the public.

I cannot help but believe that the future of the Society is as bright as, if not brighter than, it was 25 years ago, although I feel that its endeavors in the future will be more along the lines of research than they have been in the past. Along these lines I might mention the development of a new type of internal-combustion engine which will enable higher efficiencies to be obtained, so that we may utilize, instead of 3 to 5 per cent of the energy of the fuel burned, something nearer the true value of 10 to 15 per cent. This is entirely possible, but it means a considerable amount of study to accomplish the desired end.

In my opinion, the Society of Automotive Engineers will prove to be of greater value to the automotive industry in the future than it has in the past.

A. L. Riker

A Quarter-Century Retrospect

TO THOSE of us in the industry who date back into the dark ages, as it were, when there was no S. A. E. but only its progenitor, the Mechanical Branch of the A.L.A.M., the birth and continued growth of the Society have been not only a source of gratification but one of real wonder.

The old Mechanical Branch was a good beginning, but those of us who were members of it were limited to a very great degree in our activities, which were not fully understood and frequently looked at very much askance. This is not to be wondered at, as the Mechanical Branch was a distinctly commercial affair and subject to the resulting limitations.

It was only after the S.A.E. came into being that the engineer members could approach their problems in conference in a really scientific manner; and even this attitude on our part was at first hampered to some extent by a lurking suspicion—or perhaps fear is a better word—that we would "spill the beans" in those early stages of automobile development.

It is to the credit of the "mere" engineer that the Society gradually came out from under the cloud, thin though it may have been, and proved its worth to those same bosses back home. Along with this sense of fear was a lurking feeling of humor toward the boys who were trying to erect an organization that probably could not function. However, function it did, for it was founded on a real need and, once started, it just had to grow. This doesn't mean that its career has been all beer and skittles by any means. Good things don't come about as easily as that. But, having an ideal to strive toward, a loyal membership and an efficient staff, there was no stopping it.

In the case of an individual, the only reason for his existence is service in the job he finds himself filling. When service ceases or decreases, that man, and maybe the job as well, is headed for the discard. In the same way it is true that, so long, but only so long, as a society serves the industry from which it sprang will it continue to flourish. Our S.A.E. does still serve, and serve well, its parent industry, and there is no cloud in the sky affecting this fundamental of its life. It is hoped and believed that this service attitude will be fostered and strengthened by successive administrations, for therein lies its only reason for existence.

It has become a mere platitude to speak of the wonderful growth of our industry. Few, however, except those who have been in the heat of the struggle can know of the heartaches of failure, the weariness of constant strain, around and on which this growth has taken place. Many a life has been sacrificed in the struggle, and we owe all honor to those who have dropped out from time to time; all good men and true. Many others have fallen out due to failure in the struggle, in not a few cases having really given as much of value to the industry as most of us who have remained. Ultimate success is always built on a long line of failures—failures frequently as valuable as so-called successes.

So, at this time of rejoicing, I bid you all pause and remember those who started with us but are not now here. All honor to them and what they accomplished.

I often think the most unhappy man in the world and perhaps the most unfortunate is he who has no sense of humor, or one only mildly cultivated. In any walk of life such a one's path is rough, but in our hectic industry it must have been plain hell! Think what we had to struggle with back in the early '90's trying to make hot-tube ignition work, especially on a windy night. Or how the make-and-break points used frequently to break and infrequently to make. It was a nightmare, as one looks back on it, and only that deep sense of humor kept us out of insane asylums.

Then, trying to get a clutch which would disconnect and re-connect the cranky internal-combustion engine from and to the work it was supposed to do. I remember a centrifugal clutch we used at Hartford in 1904. When it behaved it was perfect. I think it was the only mechanical device I ever heard of that required the same medicine to cure both slipping and grabbing; and I claim full credit for finding that medicine. If it slipped, a small handful of Dixon's flake graphite brought it to terms; if it grabbed, the same flake graphite removed all stuttering.

In those days single-tube hosepipe tires were held to the rim by screws in metal lugs. Sometimes we used to shellac those tires to the rims to help the lugs. Did you ever try to remove one of those glued-on contraptions on a dark night by the aid of an uncertain acetylene headlight? If not, you don't and can't appreciate the present-day tire.

One could ramble on through this cob-webby past for a long time. These experiences all look funny now, but that's because of their distance. Still, we did get a lot of fun out of our jobs and I think it all centered around the element of adventure. One had to go entirely on one's own in those days if he got out of sight of the factory. The time of return was never known until one returned.

My hat is off to those old pioneers. They builded better than they knew. My hat is also off to those who have followed and improved. My best wishes go to those who will carry on and do things of which we of '95 never dreamed, and things we of 1930 do not dream now. All in all, it has been great fun and I am very gratified that I have been permitted to see it all the way through and play a little part, not only in the industry but in the S.A.E., to which, all future success.

Geo Dean

the fact that some degree of standardization must be attained to prevent the repair-parts depots in France from being swamped was realized. This led to numerous inquiries being made of the Society as to how these specifications could best be met.

Committees were appointed to consider the various features of the design and to standardize as far as possible on the different components. Conferences were held from early May until late in July, 1917, when the War Department decided that, instead of attempting to adapt existing models to meet the specifications that had been issued, a complete new vehicle that was completely standardized was to be designed. A number of the engineers who had cooperated in the earlier standardization program were called to the City of Washington to begin the work about Aug. 1. In spite of many handicaps, the design of the various parts was practically 90 per cent complete on Sept. 10, and the chassis design was finished a few days later. All


CHARLES M. MANLY
President, 1919


GEORGE W. DUNHAM
President, 1917


W. H. VANDERVOORT
President, 1915


J. G. VINCENT
President, 1920


C. F. KETTERING
President, 1918


RUSSELL HUFF
President, 1916

the details of the design were completed and checked by Oct. 1, and a week later one sample truck was completed, another being ready two days later. Only 69 days elapsed between the calling together of the engineers to begin work and the completion of the first sample vehicle.

Another triumph for the principle of standardization, which has been insisted upon and fostered by the Society as by no other organization, was the Liberty aircraft engine. This was designed by E. J. Hall and J. G. Vincent, two members of the Society, who completed the preliminary designs in approximately 48 hr. In less than a month after the design had been approved by the Aircraft Production Board, the first sample engine had been delivered to the Bureau of Standards for testing. The development work necessary to make this possible and the elaborate prepara-

DAVID BEECROFT
President, 1921

B. B. BACHMAN
President, 1922

H. M. CRANE
President, 1924

H. L. HORNING
President, 1925

T. J. LÍTLE, JR.
President, 1926

J. H. HUNT
President, 1927

tions for its production in immense quantities was conducted to a great extent by members of the Society.

The Class B truck and the Liberty aircraft engine were the two outstanding contributions of the Society to the winning of the World War. Another, which was more individual, was the active service of 562 members in the Army, the Navy or the Marine Corps of the United States or in some branch of the armed forces of the Allies. No less important a part was played by 336 others who served the Government in various civilian capacities.

S.A.E

In the fall of 1921 the Research Department was inaugurated, with Dr. H. C. Dickinson, of the Bureau of Standards, as its head. The object of this research work was to secure, through cooperative effort, accurate technical information for the use of the members of the Society and to make this information readily available. One of the first projects handled by the Department was a research on gasoline fuels, in which the American Petroleum Institute, the Bureau of Standards and the National Automobile Chamber of Commerce cooperated with the Society.

During the development of the tractor industry, special meetings on tractor design were held annually. At these, features of construction were discussed by the different engineers employed in the principal factories, and demonstrations of various models supplemented the papers presented at the technical sessions.

25TH ANNIVERSARY

More recently the scope of the work of the Society has been considerably broadened. While production and motor transportation had been discussed at the technical sessions of the Annual and Semi-Annual Meetings, their growing importance has been recognized by special meetings devoted to these two subjects. The first Production Meeting of the Society was held at Detroit on Oct. 26 and 27, 1922, and was a marked success, the total attendance being in excess of 600, with nearly 400 mem-

S. A. E.

Builded Better than They Knew

HAVING been one of the Charter Members of the American Institute of Electrical Engineers, it was only natural that when I switched from electricity to automobiles and attended the meetings of the Mechanical Branch of the A.L.A.M. I should see the necessity for, and the opportunity to, organize an Institute of Automobile Engineers. In 1903 there were not many of us, so it was a simple matter to make a canvass of the industry as it existed at that time.

Everyone approached thought it would be a good thing to have such an organization, but many were skeptical as to the necessity at that time, or the possibility of success.

Having been somewhat of an organizer of college clubs, yacht clubs and sundry other collections of individuals, which organizations uniformly died sooner or later, usually sooner, of financial insufficiency, I was loath to add another one to the list.

However, by 1904, "Constant Reader," "Veritas" and others began to write to their favorite automobile magazine about the necessity for an Association of Automobile Engineers. Most of these letters came from men who did not have access to the meetings of the A.L.A.M. Mechanical Branch. Being assured of the support of Mr. Swetland, publisher of *The Automobile*, and Mr. Ingersoll, publisher of *The Horseless Age*, I started the ball rolling, with the result that you see before you today—the only one of my club children that ever grew up.

I am asked what we hoped to accomplish and what our vision was for the future. I doubt if we had any plans for the future except to get a sufficiently numerous and influential membership so that it would be hopeless for anyone else to try to start a rival organization. An attempt at one was led by a group of high-grade chauffeurs but it soon faded away.

One of the difficulties of securing new members was that they got very little for their \$10, and of that little the less said about the quality the better. After about 200 optimists had bravely sent in their dues and persistently clamored for some return for their money, we realized that we had such a healthy infant on our hands that we must hire a professional nurse to bring him up. This is where Coker F. Clarkson enters the picture, and from then on the Society grew right along with the automobile industry, which as we all know grew faster than anything ever grew before.

Our Society has probably been of more direct and greater benefit to its affiliated industry than has any other similar organization. I believe this is abundantly proved by the prices at which cars are being sold today as compared with the prices of other manufactured commodities. The activities of the Society in disseminating knowledge and information, its unequalled system of standards, and its widespread and intimate contact with the industry have benefited the manufacturer and the buying public alike.

It is to be hoped that we shall be able to do for the aviation industry as much as we have done for the automobile industry. No one can foretell the rate or the direction of growth of aviation any more than anyone in 1903 had any vision of the automobile industry of 1930. The Society will grow in numbers and prestige just as fast as its basic industries grow. What this will be, your guess is as good as mine.

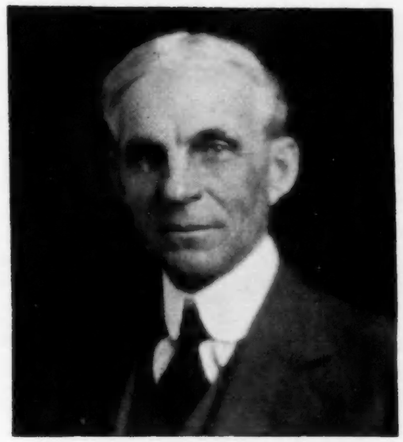
E. T. Birdsall

25TH ANNIVERSARY

bers and guests attending the Production Dinner and an average of 350 at the professional sessions.

The following spring the first Automotive Transportation Meeting was held at Cleveland, April 26 to 28. Sessions devoted to the presentation of papers dealing with the applications of motor-trucks, motorcoaches and taxicabs to transportation were held, as was also a Transportation Dinner. The keynote of this meeting was the promotion of a better understanding between the operators of steam and electric railways on one hand and the automotive industry on the other. The attendance at this meeting was 188.

Considerable attention has been paid to aviation, because intense interest in aeronautics has developed within the last few years. Numerous meetings devoted to this subject have been held in the last year in connection with aeronautical expositions in various parts of the Country or the National Air Races. The Detroit Section was the first to sponsor an Aeronautic Division, organization having been effected on Feb. 27, 1928, and to hold special meetings for members interested in aeronautics. Since that time the Chicago, Metropolitan and Southern California Sections have followed Detroit's example, and an Aeronautic Section has been formed in Wichita, Kan.



HENRY FORD
First Vice-President, 1905 to 1907

W. G. WALL
President, 1928

W. R. STRICKLAND
President, 1929

EDWARD P. WARNER
President, 1930

An intensive study of production engineering was begun by the Society in the early part of 1926, and led to the organization of the Production Division of the Standards Committee. The function of this Division was the formulation of standards and the outlining of recommended practices pertaining to machine tools, production methods and practices and several other projects relative to the manufacture of automotive apparatus and materials. These production standards and recommended practices differ from the Society's design standards and recommended practices in that the latter pertain to the product itself, whereas the former relate only to its manufacture.

At various times prior to 1928 demands for specific representation, on the Council, of such fields or phases of automotive engineering as fleet operation, maintenance and service, body engineering and production engineering had been made by members interested in those activities. The question was discussed at some length at the 1928 Semi-Annual Meeting, and this discussion led to the appointment of a Reorganization Committee to study the whole question and report its recommendations at the 1929 Annual Meeting.

After a number of conferences the Committee devised a plan by which the various automotive engineering professional activities were accorded representation on the Council and by Vice-Presidents. This was presented at the 1929 Annual Meeting and discussed at some length. As outlined by the Committee, professional activities, such as Aircraft Engineering, Aircraft-Engine Engineering, Diesel-Engine Engineering, Motor-Truck and Motorcoach Engineering, Passenger-Car Engineering, Passenger-Car Body Engineering, Production Engineering and Transportation Engineering, were to be recognized by the Council and the Society.

The necessary amendments to the Constitution and changes in the By-Laws were proposed at that meeting and the former were adopted by the voting members of the Society in the summer of 1929, the Council subsequently changing the By-Laws to conform to the plan and put it into effect at the beginning of this year, which has made the Society procedure more effective and more closely knit.

COKER F. CLARKSON

Guiding Spirit of the Society
for 21 Years as Its
Secretary and General Manager

The Pilot of the S. A. E.

FOR THE SOCIETY to celebrate its 25th anniversary and rejoice in the progress made without giving recognition to him who, more than anyone else, has been responsible for its growth and success would indicate a lack of appreciation that certainly does not exist. Those officers and members who have been identified with the activities of the Society from its infancy are keenly aware of what it has meant to have at the controls a man in whose sound judgment, far-sightedness and loyalty absolute confidence could be placed.

For the 20 years since 1910, Coker F. Clarkson has served as Secretary and General Manager of the Society and in that period has labored unceasingly day and night for the advancement and prestige of the organization. He has won the high regard and affection of all who have worked with and under him and general recognition among the membership of his capabilities and dependability as a Manager and a friend. The sympathy of all is with him in the illness that has confined him to his home for the last four months, and all are wishing for and anticipating his complete recovery and early return to his accustomed activities.

Throughout the score of years in which he has carried the growing burden of the affairs of the Society, he has been tactful, self-effacing, square-dealing and friendly with all, deferring to the views and desires of the officers and Councilors and always ready to be helpful to the successive incumbents of office.

So extensive and manifold have been Mr. Clarkson's duties with the Society that no adequate review of them is possible; it may truthfully be said that in the earlier years of his pilotage he *was* the Society. He navigated it through the uncertain winds of the early part of its long non-stop flight to assured success. Perhaps most of the crew and passengers aboard in the latter part of the voyage are not familiar with the personal history of the pilot which will be of undoubted interest to them.

Mr. Clarkson was born at Des Moines, Iowa, in 1870, and was named for his grandfather, Coker Fifield Clarkson, whose paternal grandfather came to America in 1779 and settled in New Hampshire. In 1870 the present Mr. Clarkson's grandfather and his two sons purchased the *Iowa State Register*, in Des Moines, and Grandfather Clarkson served for 20 years as agricultural editor, being one of the pioneers in agricultural education in Iowa. One of the sons, James S. Clarkson, was the father of the Secretary of the S.A.E., who entered the State University of Iowa at the age of 14 years and was graduated from the Phillips Exeter Academy in 1888. The following year he was in the Government service in the Post Office Department, and in 1894 was graduated from Harvard University with the degree of Bachelor of Arts and with first rank in logic. Two years later he was graduated from the Harvard School of Law and was admitted to the bar in Philadelphia.

During 1897 and 1898, Mr. Clarkson was engaged in connection with the installation of an underground telephone system in Philadelphia, after which he spent several years in New York City in work on technical, legal, patent, laboratory and automobile subjects. From 1905 to 1909 he was secretary of the Mechanical Branch of the Association of Licensed Automobile Manufacturers and publicity manager and assistant general manager of the A.L.A.M. During that period he was editor of the Mechanical Branch Bulletins and of the A.L.A.M. weekly digest of current technical literature relating to automobile subjects. He also edited four annual volumes of the Handbook of Gasoline Automobiles, the publication of which annual has been continued successively to the present time by the A.L.A.M., the Automobile Board of Trade and the National Automobile Chamber of Commerce.

With this background of connection and experience with the standardization and other technical work of the Mechanical Branch of the A.L.A.M., Mr. Clarkson was the logical choice when the time came for the S.A.E. to select a man qualified for the position of Secretary and General Manager. It is also obvious that he would take a keen interest in editorial work, which he has maintained to the present time. For several years after the S.A.E. BULLETIN was started in 1911, he did all of the editorial work and even after THE BULLETIN was succeeded by the JOURNAL OF THE SOCIETY in 1917 continued to do a large part of it.

During the World War period Mr. Clarkson was associated with the Council of National Defence, of which Howard E. Coffin was chairman; was a member of the United States War Industries Board, of which his brother Grosvenor Clarkson was secretary; and was a member of the International Aircraft Standards Board.

In addition to the writing of innumerable reports of meetings of the Society and of the Council and various committees, Secretary Clarkson has written and presented numerous addresses at Society meetings, among them being the following, which show his predominating interest in standardization:

Possibilities of Tractor Standardization, published in THE BULLETIN, March, 1917, p. 637.

International Aircraft Conference, published in TRANSACTIONS, 1918, part 2, p. 100.

The Place of Sections in Society Activities, THE JOURNAL, March, 1920, p. 177; and TRANSACTIONS, 1920, part 1, p. 917.

Chicago Aeronautic Meeting Address, THE JOURNAL, April 20, 1920, p. 270; and TRANSACTIONS, 1920, part 1, p. 463.

Automotive Engineering Standardization and Progress, THE JOURNAL, October 1920, p. 356.

Standardization in the Automobile and in the Aeronautic Industry, The S.A.E. JOURNAL, January, 1929, p. 30.

In the words of Past-Treasurer Whittelsey:

> As we pause to review the activities of the S.A.E. during its first quarter of a century, we see grouped about us the men who were and are the pioneers of this greatest industry of the entire world. Throughout this entire span of years there has been a guiding spirit to bring about the great achievements of our industry; a rugged hand, tempered by a sympathetic heart, a trained far-seeing mind; one who has understood human nature so well that he could extract the best from others and blend them into an ever-increasing fellowship which has mastered the economic problem and has been the means of bringing the industry to the high plane on which it flourishes today. All through these years we see this guiding spirit in Coker F. Clarkson, as the Secretary and General Manager of the S.A.E.

History of Automotive Standardization

Beginning 30 Years Ago, It Has Been a Powerful Factor in the Amazing Progress of the Industry

STANDARDIZATION has today become one of the major factors in economic engineering, purchasing and manufacturing and is to a large extent responsible for the tremendous advance the automobile industry has made during the past 25 years. The spirit of voluntary cooperation among the engineers and executives of virtually all the manufacturers of materials, parts and complete vehicles has made this standardization possible and has possibly been more pronounced in the automobile industry than in any other.

Records show that in the early days of the industry there was a determination to develop standardization to the point where it was logical and useful but to avoid over-standardization. There were some then, as now, who lacked interest in this work or did not give it their full support, but on the whole the entire industry has realized what a tremendous help the pooling of ideas and experience as expressed in standardization has been for the common good.

First Automobile Standards

On Dec. 1,1900, the National Association of Automobile Manufacturers was formed in New York City to foster the interests of the automobile manufacturers, and in 1902 the N.A.A.M. adopted the first automobile standards. These were for spacing of tire lugs and holes, rim sections and lamp brackets. In 1904 the accessory members of the N.A.A.M. formed the Motor and Accessories Manufacturers, Inc., which in 1929 merged with the Automotive Equipment Association to form the Motor and Equipment Association. In March, 1903, the A.L.A.M. was organized by the automobile manufacturers licensed under the Selden patent, and in February, 1905, the American Motor Car Manufacturers Association was formed by the independent manufacturers but this was succeeded in February, 1910, by the Automobile Board of Trade. In 1913 the N.A.A.M. and the Automobile Board of Trade were combined into the National Automobile Chamber of Commerce and continued to support the standardization work of the S.A.E. that had been taken over from the Mechanical Branch of the A.L.A.M. in 1910.

The Mechanical Branch of the A.L.A.M. was organized in 1905 to study and discuss the designs and construction of foreign and domestic automobiles and the new ideas that were rapidly being developed in the primitive designs of those days. The activities of the Mechanical Branch paved the way to the mutual effort and confidence of the manufacturers and their engineers that proved such a solid foundation for the standardization activities in the industry during the years that followed.

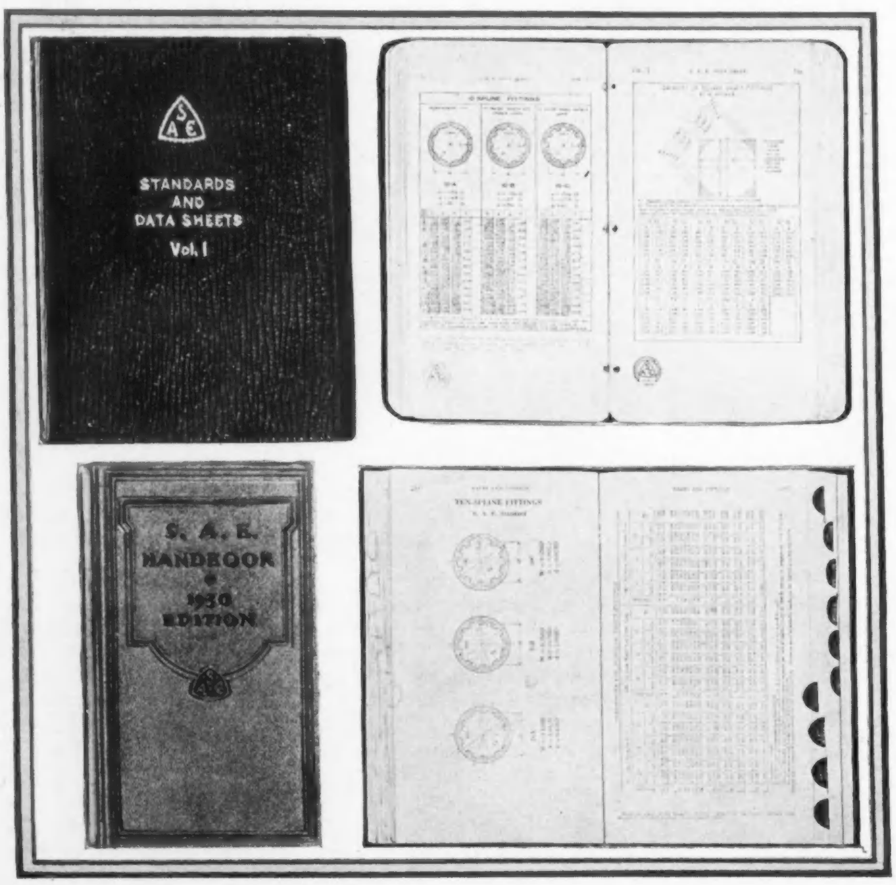
A.L.A.M. Mechanical-Branch Work

At the meeting of the Mechanical Branch of the A.L.A.M. in New York City on July 7, 1905, A. L. Riker said: "It would be a great plan to get at some definite names for the various parts of the automobile, because, as we all know, not every manufacturer and dealer calls the same parts by the same name."

It was not until June, 1911, however, that the first report on Nomenclature was prepared.

Committees of the Mechanical Branch were appointed to study various technical subjects and reported on them from time to time for general discussion by the members and for their confidential information. Although many of these reports later were the foundation for S.A.E. Standards, they were not at first adopted as standards by the A.L.A.M. Among these reports were analyses and physical tests of aluminum, bronze, brass and other nonferrous alloys and iron and steels, lubricating oils, steel tubing, tires, screws, bolts and nuts, and spark-plugs. The first A.L.A.M. standard specification was for Screws, Bolts and Nuts, which was adopted by the Mechanical Branch on Aug. 6, 1906. This was followed by A.L.A.M. Standards for Spark-Plugs, Cotter-Pins, Steel Tubing and a number of other items.

The idea of having a National organization of the automobile engineers was discussed first by E. T. Birdsall, H. M. Swetland and A. H. Whiting in 1904 and culminated in organizing the



FIRST VOLUME OF S.A.E. STANDARDS AND DATA SHEETS AND LATEST VOLUME OF THE S.A.E. HANDBOOK

S.A.E. in January, 1905. E. T. Birdsall served as Secretary until Alexander Churchward was appointed in 1908. The Society was incorporated in 1909, and in 1910 Coker F. Clarkson was appointed Secretary and General Manager and has ably filled these positions ever since.

S.A.E. Standards Committees

Several years following the organization of the Society, W. P. Kennedy suggested at the meeting of the Society in July, 1910, that a committee be organized to develop standards, and at this meeting the Council appointed the first Standards Committee in the automotive industry. It consisted of 16 Divisions, with the following members:

1910 STANDARDS COMMITTEE

Henry Souther, *Chairman*

ALUMINUM AND COPPER-ALLOYS DIVISION

William H. Barr, J. J. Aull
F. W. Cooke, George W. Dunham
Thomas J. Fay, E. S. Fretz
George M. Holley, S. P. Wetherill, Jr.
H. W. Gillett

BALL AND ROLLER-BEARINGS DIVISION

D. F. Graham, A. P. Sloan, Jr.
H. W. Alden, W. A. Frederick
David Fergusson, Henry Hess
W. P. Kennedy, Elwood Haynes
A. L. Riker, Howard Marmon
S. P. Wetherill, Jr.

BROACHES DIVISION

C. E. Davis, C. W. Spicer
F. L. Eberhardt, G. E. Merryweather

GEAR-METAL CONSTANTS DIVISION

G. W. Sargent, C. H. Taylor
J. M. Mack, H. W. Alden
Henry Hess, W. H. VanDervoort

CARBURETER DIVISION

G. G. Behn, A. L. Riker
George M. Holley, J. G. Sterling
H. P. Maxim, Howard Marmon

FRAME SECTIONS DIVISION

James H. Foster, W. H. VanDervoort
A. L. Riker, J. G. Perrin
L. R. Smith, W. P. Kennedy

IRON AND STEEL DIVISION

W. P. Barba, A. R. Gormully
Elwood Haynes, E. L. French
S. V. Hunnings, Arthur Holmes
M. T. Lothrop, Russell Huff
George L. Norris
Joseph Schaeffers

LOCK-WASHER DIVISION

Charles T. Jeffery
Henry A. Bugie
A. C. Bergmann
Frederick S. Sayre

MISCELLANEOUS DIVISION

H. P. Maxim, G. G. Behn
J. C. Chase, S. P. Wetherill, Jr.

NOMENCLATURE DIVISION

P. M. Heldt, C. H. Taylor
A. L. McMurtry, A. H. Whiting

GASOLINE MOTOR CHARACTERISTICS RECORD DIVISION

H. G. Chatain, E. T. Birdsall
B. D. Gray, Alex. Churchward

SEAMLESS STEEL-TUBES DIVISION

H. W. White, H. W. Alden
W. H. Tuthill, J. Jay Dunn
W. S. Gorton, C. E. Reddig

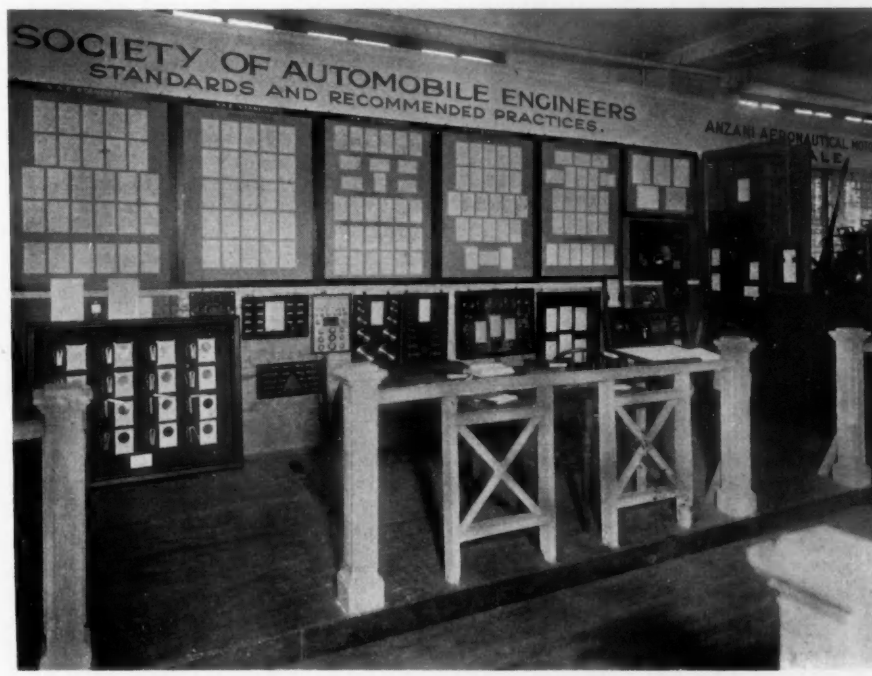


EXHIBIT OF S.A.E. STANDARDS AT THE FIRST NATIONAL AERONAUTIC SHOW IN NEW YORK CITY IN FEBRUARY, 1912

SHEET METALS DIVISION

Robert Skemp, C. E. Whitney
L. R. Smith, F. C. Burkhardt
James H. Foster, C. E. Lozier
A. R. Gormully

SPRINGS DIVISION

A. C. Bergmann, W. H. Tuthill
G. A. Weidely, Charles T. Jeffery
Christian Girl, George S. Case
W. H. Son

TIRE EFFICIENCY DIVISION

F. J. Newman, David Fergusson
Hermann F. Cuntz, Bruce Ford
H. W. Alden

WOOD-WHEEL DIMENSIONS AND FASTENINGS FOR SOLID TIRES

W. P. Kennedy, Charles L. Schwarz
J. M. Mack, Charles B. Whittelsey
Clarence B. Hayes

The Standards Committees were appointed annually thereafter at the beginning of each administrative year. During the first years of the Standards Committees, each Division undertook a rather closely defined project, but as time went on the activities of the Standards Committee grew and each Division broadened the scope of subjects on which it worked. New Divisions were organized from time to time during this expansion and, in the case of projects that did not prove practicable or that could be consolidated, some of the Divisions were discontinued. The chart following p. 702 is a chronology of the officers of the Standards Committee and of the Divisions and their respective Chairmen since the first Standards Committee was organized.

The standards work of the Society progressed rapidly from the start under the able guidance of Henry Souther, the first Chairman of the Standards Committee, who was succeeded in 1915 by K. W. Zimmerschied. Shortly thereafter, an office was opened in Detroit for the Standards Committee activities, where they could be under Mr. Zimmerschied's supervision during his chairmanship.

The S.A.E. Handbook

At the meeting of the Society in July, 1910, H. G. McComb proposed that the automobile engineering information sheets that were issued by the Society be published in a loose-leaf data book.

S. A. E.

The suggestion was adopted and in June, 1911, at the meeting of the Society at Dayton, Ohio, it was decided that the A.L.A.M. Standards that were adopted by the Society should be renamed and published as S.A.E. Standards.

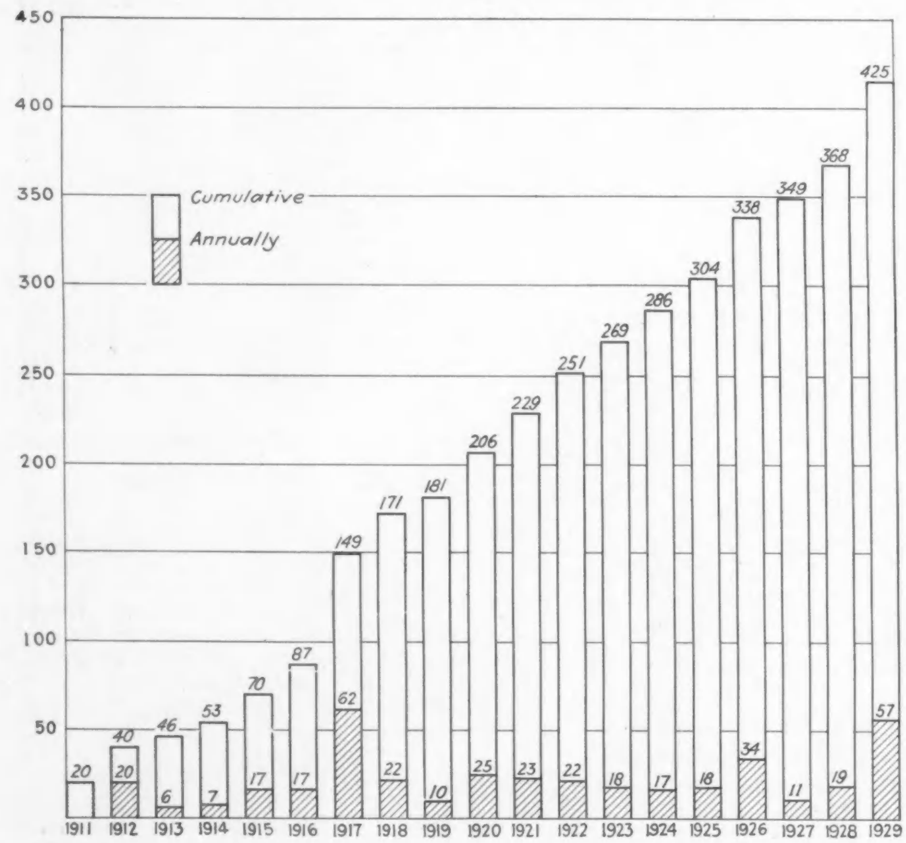
Among the subjects reported for adoption at that time were Iron and Steel Specifications; Non-Ferrous Metals Specifications; Lock-Washers; Screws, Bolts and Nuts and Screw-Threads; Broaches; Carbureters and Carbureter-Tube Fittings; Yoke and Rod Ends; Spark-Plug Tolerances; and Sheet Metals.

Shortly after the meeting in June, 1911, the first issue of the S.A.E. DATA BOOK, which later became the S.A.E. HANDBOOK, was issued and contained the standards that had been adopted by the Society and other general engineering data that were considered of permanent use to automobile engineers. Revised and new standards adopted in succeeding years by the Society were printed on the loose sheets and distributed to the members following each meeting of the Standards Committee. In 1913 the sheets were divided into two volumes, Vol. I becoming the S.A.E. HANDBOOK and containing in the main the standards and recommended practices of the Society and Vol. II being the S.A.E. DATA BOOK containing general engineering data.

Growth of Standards Handbook

In 1920, the DATA SHEETS were discontinued because a large percentage of the data were out of date and most of them were obtainable from other sources. Vol. I continued to grow and soon became unwieldy; moreover, much trouble was experienced by the members of the Society in keeping their books up-to-date. Therefore the Council decided that the S.A.E. HANDBOOK should be published as a bound volume and the first number in this form was issued in March, 1926. The 1930 edition embodies the improvements that have made the S.A.E. HANDBOOK a modern practical publication containing nearly 450 engineering Standards and Recommended Practices of the Society, classified as follows, and other valuable matter recognized to be of importance to automotive engineers.

Class	*Number*
Aeronautics	37
Powerplants	36
Transmissions	25
Electrical Equipment	82
Parts and Fittings	68
Screws and Screw-Threads	56
Tires, Rims and Wheels	18
Chassis and Body	24
Marine	10
Production	6
Miscellaneous	10
Materials	65
Nomenclature	1 (19 Sections)



APPROXIMATE NUMBER OF STANDARDS AND RECOMMENDED PRACTICES ADOPTED BY THE SOCIETY, BY YEARS

S.A.E. Production Standards

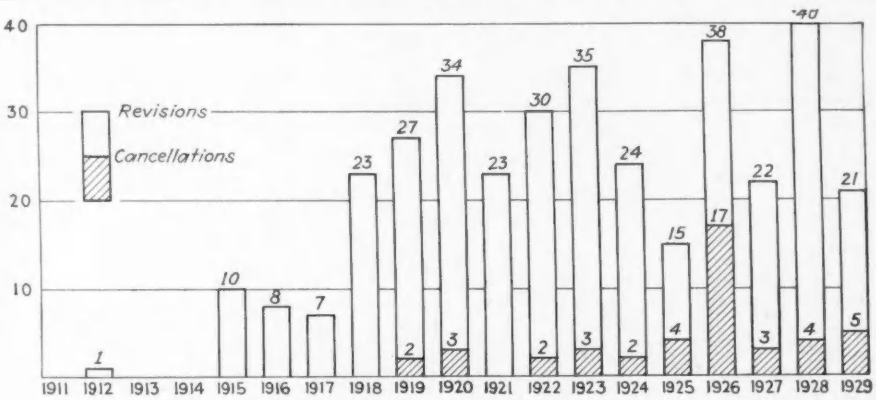
Virtually all of the Standards and Recommended Practices published in the S.A.E. HANDBOOK relate to the product rather than to its manufacture. The Production Engineering Standards, the first of which was adopted in January, 1927, relate to the tools, equipment, methods or processes for manufacturing the product and are published separately in standard 8½x11-in. pamphlets and by reference in the S.A.E. HANDBOOK. This form of publication was adopted for several reasons and in accordance with the wishes of the members of the Society directly engaged in production engineering at the time these standards were first adopted. Each edition of the S.A.E. HANDBOOK is sent to all members of the Society and the Production Standards are sent to those who desire them.

It is probable that as distinctively Diesel-Engine Standards are adopted by the Society, they will be published in a Diesel Section of the S.A.E. HANDBOOK, as many of the existing S.A.E. Standards are applicable to this type of engine and all Diesel Standards should be issued in one volume.

Scope of S.A.E. Standardization

The Society's standardization work during the first six years was devoted almost entirely to the automobile industry, although many of the standards that were adopted, such as the Iron and Steel and the Non-Ferrous Metal Specifications, became widely known and were used in many other mechanical industries. During these years there were also organized other National groups concerned with gas and gasoline-powerplant engineering. These were the American Society of Aeronautic Engineers, the Society of Tractor Engineers, the National Gas Engine Association and the National Association of Engine and Boat Manufacturers. In 1916 the two societies named were merged with the Society and the following year the engineering activities of the two associations were consolidated with the S.A.E. It was at this time that the name of the Society

25TH ANNIVERSARY



APPROXIMATE NUMBER OF REVISIONS AND CANCELLATIONS OF S.A.E. STANDARDS AND RECOMMENDED PRACTICES, BY YEARS

was changed to the Society of Automotive Engineers, Inc.

The Aeronautic-Engine Division of the Standards Committee was organized in 1916 and continued in 1917 as the Aeronautic Division. The Marine Division and the Tractor Division were organized, and revised Standards Committee Regulations governing the organization and procedure of the Committee were adopted in the latter year largely because of the rapid expansion of the scope and work of the Committee.

Military Standardization Work

Immediately following the entry of the United States into the World War, the Aeronautic Division became very active in developing standards and specifications for parts and materials intended primarily for use in military airplanes in accordance with the program of the Aeronautic Specifications Section of the Bureau of Construction and Repair, United States Navy Department. Among the Division's reports approved at the Summer Meeting in 1917 were those on Loops and Ferrules, Cable Ends, Stick Controls, Thimbles for Wire Ends, Turnbuckles, Marking of Pipe Lines, Engine Supports, Shackles, Clip Ends, Rod-End Pins, Airplane Bolts and Nuts, and Magneto Dimensions. This work continued as one of the main activities of the Standards Committee during the war period. The Aeronautic Division and the Standards Committee also cooperated with the United States Army Air Corps, which at that time was drafting similar specifications for the Army Air Service. The Society also cooperated in developing the standardized Class B Army truck and other equipment used in the Motor Transport Corps and the Ordnance Department.

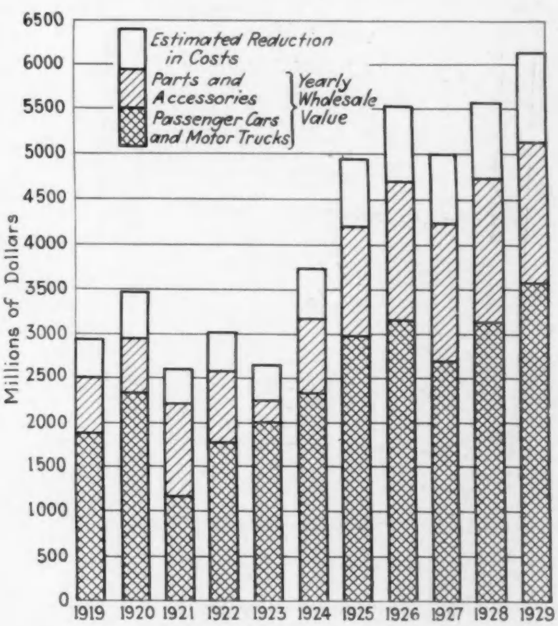
The first international automotive engineering standard, formulated by the Society in cooperation with the British, French and Italians, was adopted in 1917 for interchangeable aeronautic spark-plugs for military service. A number of other international standards were in process of formulation but were not completed due to the cessation of the World War.

Production and Diesel-Engine Divisions

For several years the Society had been extending its work into production engineering in the automotive industry, and in January, 1926, the Production Division of the Standards Committee was organized to formulate standards relating to manufacture of the product instead of the product itself. Although only a few production engineering standards have so far been adopted and published by the Society, a number of very important ones are in progress under the procedure of the American Standards Association, in which the Production Division is cooperating. The most recent addition to the family of Divisions of the Standards Committee is the Diesel-Engine Division. For several years Diesel-engine representatives have served on the Engine Division, which is now the Gasoline-Engine Division, so as to maintain contact with this field of engineering. In 1929 it became evident that the Diesel engineers felt that the time had arrived when standards for Diesel engines could be developed, and accordingly the Diesel-Engine Division was organized at the time the 1930 Standards Committee was appointed. Although this Division has only started its work, several existing S.A.E. Standards will be extended and approved for this type of engines and a number of distinctly Diesel specifications are now in process of formulation.

S.A.E. Standards Exhibits

The first prepared exhibit of S.A.E. Standards and Recommended Practices was displayed at the Automobile Show held under the auspices of the Automobile Board of Trade at the old Madison Square Garden in New York City in June, 1912. This attracted so much attention that later it was decided to have similar exhibits at the National Automotive Shows; consequently panels were made up showing such S.A.E. Standards as Carbureter Fittings, Rod and Yoke Ends, Screws, Bolts and Nuts, Brake-Lining, Broaches, Lock-Washers, S.A.E. Steel Physical-Test Specimens, S.A.E. Screw-Threads, Electric Wire and Cable, Incandescent Lamps, Ball-Bearings, and many other suitable items. These, together with additional exhibits, were displayed in the Society's booths at several of the National Tractor Shows, the National Motorboat Shows, at most of the National Aircraft Shows during the last two years and also at some of the National exhibitions held by other organizations such as the National



REDUCTION OF COST OF AMERICAN AUTOMOTIVE PRODUCTS EFFECTED BY S.A.E. STANDARDIZATION

Based on a Survey in 1921 Among 146 Executives and Engineers, Which Indicated a 15-Per Cent Average Reduction for the Industry in General, and the Yearly Wholesale Values Given in Facts and Figures Published by the National Automobile Chamber of Commerce

Automobile Chamber of Commerce. Much information about the S.A.E. Standards work was promulgated in this way and these exhibits have without doubt been helpful in popularizing standardization in all branches of the automotive industry.

National and International Standardization

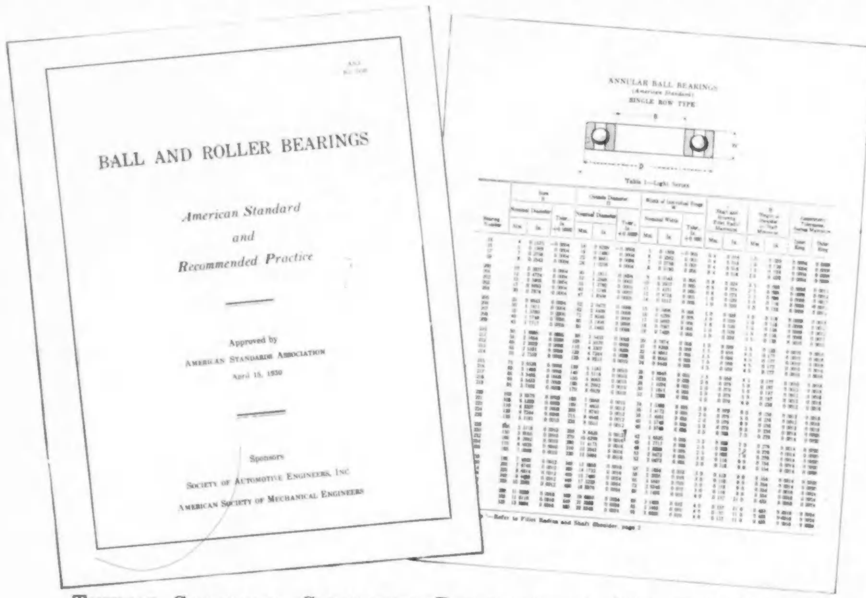
Standardization had been practised both in the United States and abroad in the automotive and other industries for many years to a limited extent, but the World War brought a keener realization everywhere of the value of standardization and efforts were soon made to bring various industrial groups together for National standardization.

American Standards Association

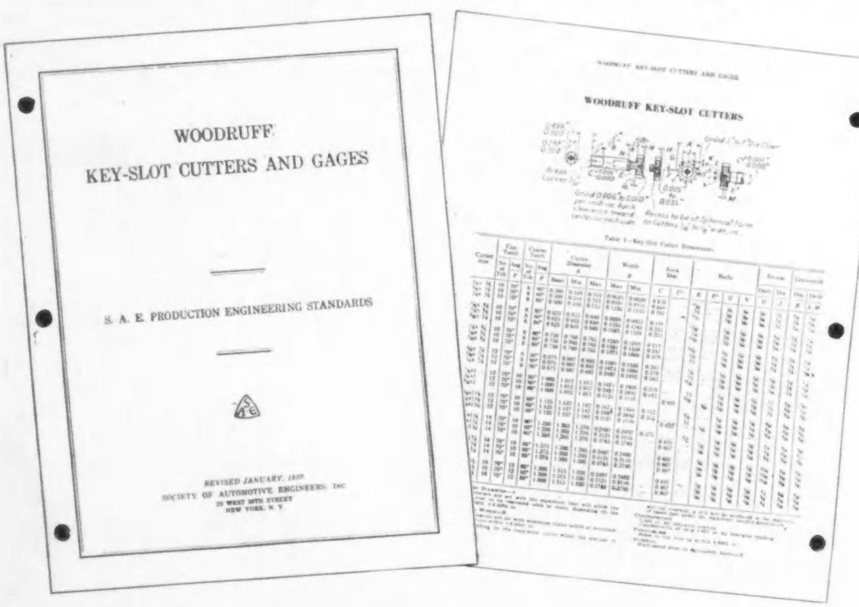
Several such groups had already been formed in Europe when the American Engineering Standards Committee was organized in 1918 as a National agency or clearing house for American industrial standardization, its membership comprising National engineering societies, trade associations and a number of Federal Bureaus and Departments. Sectional Committees for individual standardization projects were organized under the procedure of the A.E.S.C., one of the first being sponsored by the S.A.E. and the American Society of Mechanical Engineers to review the first report of the National Screw-Thread Commission. With the adoption of the American Standard for Screw-Threads, other projects followed, among which were standardization of bolt, nut and rivet proportions, ball-bearings, small tools and machine-tool elements, and a number of safety codes for various classes of manufacturing machinery, all for general American practice.

As the American Engineering Standards Committee progressed, it was felt that the scope of its work should be broadened beyond industrial engineering projects and accordingly, in November, 1928, the A.E.S.C. was reorganized as the American Standards Association under a revised Constitution.

In 1914 the Ball and Roller-Bearings Division of the S.A.E. Standards Committee was working on international standardization of ball-bearings, but this was indefinitely delayed by the World War. In 1918 a Sectional Committee on the Standardization of Ball-Bearings was organized under the sponsorship of the Society and the A.S.M.E., primarily for establishing international standardization of various types of ball-bearing. Although these bearings had originated in Europe, a number of differences between European and American practice had developed that prevented their international interchangeability; but through this Sectional Committee's work, which was based on the standards originally adopted by the Society in 1911, a proposal for radial ball-bearings of the single-row type that was in practical agreement in all countries concerned was finally approved by the American Standards Association in April, 1930.



TYPICAL SECTIONAL COMMITTEE REPORT OF AN AMERICAN STANDARD



LOOSE-LEAF FORM IN WHICH PRODUCTION-ENGINEERING STANDARDS ARE ISSUED BY THE SOCIETY

It is interesting to note that a number of S.A.E. Standards have been adopted in several of the principal European countries, either in their present form or by conversion of the inch standard dimensions to metric equivalents, and these undoubtedly will become well established international standards.

At the beginning of the 1930 administrative year of the Society, it was a sponsor for 10 Sectional Committees and was represented on 17 others for projects under the procedure of the American Standards Association. The Society also had representatives on more than 30 Committees of other National organizations and Government Departments and Bureaus, all of which are working on projects

Personnel of Standards Committee from 1910 to 1930,

	1910	1911	1912	1913	1914
CHAIRMEN	Henry Souther	Henry Souther	Henry Souther	Henry Souther	Henry Souther
VICE-CHAIRMEN					
Agricultural Power Equipment					
Aircraft					
Aircraft Engine					
Axle and Wheel					
Ball and Roller Bearings	David Fergusson	David Fergusson	David Fergusson	David Fergusson	Howard Marmon
Carbureters	G. G. Behn	G. G. Behn	G. G. Behn	G. G. Behn	G. G. Behn
Chain					
Commercial Car Wheels				W. P. Kennedy	W. P. Kennedy
Data Sheet				B. D. Gray	B. D. Gray
Diesel Engine					
Electrical Equipment				A. L. Riker	A. L. Riker
Electric Transportation					
Electric Vehicle					A. J. Slade
Foreign Cooperation					
Frames					
Frame Sections	J. H. Foster	J. G. Perrin	J. G. Perrin	J. G. Perrin	J. G. Perrin
Gasoline Engine					
Gear-Tooth Shapes			Henry Hess	Henry Hess	
Iron and Steel	Henry Souther	Henry Souther	Henry Souther	Henry Souther	Henry Souther
Isolated Electric Lighting Plant					
Joint Standards			Henry Hess	Henry Hess	
Lighting					
Lock Washers	J. E. Wilson	J. E. Wilson	J. E. Wilson	J. E. Wilson	J. E. Wilson
Lubricants					
Motorboat and Marine Engine					
Motorcoach					
Motorcoach and Motor-Truck					
Motorcycle					
Motor Testing				J. L. Heinze	J. L. Heinze
Motor-Truck			W. P. Kennedy	W. P. Kennedy	W. P. Kennedy
Nomenclature	P. M. Heldt	P. M. Heldt	H. E. Coffin	E. J. Stoddard	A. B. Cumner
Non-Ferrous Metals	Wm. H. Barr	Wm. H. Barr	Wm. H. Barr	Wm. H. Barr	Wm. H. Barr
Parts and Fittings	A. L. Riker	A. C. Bergmann	A. L. Riker	Arthur Holmes	J. G. Utz
Passenger-Car					
Passenger-Car Body					
Pleasure-Car Wheels				Henry Souther	Henry Souther
Production					
Radiator					
Research				D. L. Gallup	D. L. Gallup
Screw-Threads					
Seamless Steel Tubes	H. W. Alden	H. W. Alden	H. W. Alden	H. W. Alden	H. W. Alden
Shaft Fittings	C. E. Davis	C. E. Davis	C. W. Spicer	C. W. Spicer	C. W. Spicer
Sheet Metals	Robt. Skemp	T. V. Buckwalter	T. V. Buckwalter	T. V. Buckwalter	
Springs	A. C. Bergmann	A. C. Bergmann	H. L. Pope	H. L. Pope	H. L. Pope
Stationary Engine					
Storage-Battery					
Technical Index			Henry Hess	Henry Hess	
Tire and Rim					
Transmission					
Wheel Dimensions	W. P. Kennedy	W. P. Kennedy	W. P. Kennedy		

Showing Successive Chairmen and Vice-Chairmen of Committee, Divisions

1915	1916	1917	1918	1919	1920	1921
K. W. Zimmerschied	A. L. Clayden	J. G. Utz	B. B. Bachman	B. B. Bachman	B. B. Bachman	B. B, Bachr
						W. R. Strick W. A. Chrys
				DIVISIONS APPOINTED ANNUALLY		
		H. L. Horning	Dent Parrett	Dent Parrett	E. A. Johnston	E. A. Johnst
		C. M. Manly	C. M. Manly	C. M. Manly	H. M. Crane	H. M. Cran
	Henry Souther					
						G. W. Dunh
F. G. Hughes	F. G. Hughes	F. G. Hughes	F. G. Hughes	F. G. Hughes	W. R. Strickland	W. R. Stricl
J. J. Aull	J. J. Aull					
F. L. Morse	F. L. Morse	F. L. Morse	L. M. Wainwright	L. M. Wainwright	L. M. Wainwright A. L. Clayden	W. F. Cole
		A. C. Bergmann	P. M. Heldt			
A. L. Riker	A. L. Riker	W. A. Frederick	W. A. Chryst	W. A. Chryst	A. H. Timmerman	A. D. T. Lil
					E. R. Whitney	
A. J. Slade	A. J. Slade	Bruce Ford	Bruce Ford	No. Div. Apptd.	No Div. Apptd.	E. L. Clark
A. L. Clayden	A. L. Clayden	A. L. Clayden	A. L. Clayden			
					E. A. DeWaters	E. A. DeWa
J. G. Perrin						
	W. T. Fishleigh	W. T. Fishleigh	W. A. Frederick	C. C. Hinkley	H. C. Snow	H. C. Snow J. B. Fisher
K. W. Zimmerschied	K. W. Zimmerschied	K. W. Zimmerschied	H. L. Greene	F. P. Gilligan	F. P. Gilligan	F. P. Gillig
					L. S. Keilholtz	L. S. Keilho
		W. E. McKechnie	M. W. Hanks	H. M. Crane	H. M. Crane	C. E. Godle
C. H. Loutrel	C. H. Loutrel					
			H. L. Horning	No. Div. Apptd.	A. P. Eves	H. C. Mou
		H. R. Sutphen	H. R. Sutphen	Jos. VanBlerck	Jos. VanBlerck	Jos. Van Bl
			W. S. Harley	W. S. Harley	W. S. Harley	W. S. Harl
W. P. Kennedy	W. P. Kennedy	H. D. Church	B. B. Bachman	L. P. Kalb	L. P. Kalb	A. K. Brun
	K. W. Zimmerschied	K. W. Zimmerschied	Herbert Chase	No. Div. Apptd.	No Div. Apptd.	No Div. Ap
			C. M. Manly	C. M. Manly	J. J. Aull	J. J. Aull
J. G. Utz	J. G. Utz	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	Clarence C
						Ralph Mur
						E. G. Budd
E. R. Hall						
					K. F. Walker	J. D. Harri
D. L. Gallup	D. L. Gallup	D. L. Gallup				
						E. H. Ehrr
			C. W. Spicer	C. W. Spicer	C. W. Spicer	Consolidat Parts and I
C. W. McKinley	C. W. McKinley	C. W. McKinley	C. W. McKinley	W. M. Newkirk	W. M. Newkirk	R. A. Scha
			Chas. Kratsch	L. S. Keilholtz	H. N. Edens	T. C. Men
		G. E. Goddard				*Bruce For
		K. W. Zimmerschied	C. B. Whittelsey	C. B. Whittelsey	S. P. Thacher	J. G. Vinc
		A. W. Copland	A. W. Copland	A. W. Copland	A. W. Copland	A. W. Cop

visions Organized and Year of Their Organization, and Successive Chairmen of

1921	1922	1923	1924	1925	1926	1927
B. B, Bachman	E. A. Johnston	E. A. Johnston	E. A. Johnston	E. A. Johnston	F. A. Whitten	K. L. Herrmann
W. R. Strickland W. A. Chryst	W. G. Wall C. M. Manly	C. M. Manly G. W. Dunham	C. M. Manly C. C. Carlton	C. M. Manly C. C. Carlton	K. L. Herrmann R. S. Begg	C. M. Manly A J. Scaife
E. A. Johnston	John Mainland	C. B. Rose	J. F. M. Patitz	J. F. M. Patitz	O. W. Sjogren	G. A. Young
H. M. Crane	H. M. Crane	H. M. Crane	H. M. Crane	H. M. Crane	E. P. Warner	E. P. Warner
G. W. Dunham	G. W. Dunham	C. C. Carlton	C. C. Carlton	C. C. Carlton	C. C. Carlton	L. R. Buckendale
W. R. Strickland	F. W. Gurney	F. W. Gurney	F. W. Gurney	F. W. Gurney	H. E. Brunner	H. E. Brunner
W. F. Cole	W. F. Cole	H. S. Pierce	H. S. Pierce	H. S. Pierce	H. S. Pierce	Discontinued
A. D. T. Libby	F. W. Andrew	F. W. Andrew	F. W. Andrew	T. L. Lee	T. L. Lee	B. M. Leece
E. L. Clark	E. L. Clark	E. L. Clark	J. G. Carroll	J. G. Carroll	C. R. Skinner, Jr.	C. R. Skinner, Jr.
E. A. DeWaters	E. V. Rippingille	Wm. A. McKinley	C. C. Bowman	C. C. Bowman	Discontinued	
H. C. Snow J. B. Fisher	J. B. Fisher	J. B. Fisher	R. J. Broege	W. C. Ware	A. F. Milbrath	A. F. Milbrath
F. P. Gilligan	F. P. Gilligan	F. P. Gilligan	F. P. Gilligan	J. M. Watson	J. M. Watson	J. M. Watson
L. S. Keilholtz	E. B. Newill	E. B. Newill	F. L. Tubbs	F. L. Tubbs	C. B. Jahnke	C. B. Jahnke
C. E. Godley	W. A. McKay	W. A. McKay	C. A. Michel	C. A. Michel	C. A. Michel	C. A. Michel
H. C. Mougey	H. C. Mougey	H. C. Mougey	H. C. Mougey	H. C. Mougey	H. C. Mougey	W. S. James
Jos. Van Blerck	Jos. VanBlerck		J. W. Hussey	C. A. Carlson	C. A. Carlson	C. A. Carlson
				G. A. Green	A. J. Scaife	A. J. Scaife
W. S. Harley	C. B. Franklin	C. B. Franklin	C. B. Franklin	A. W. S. Herrington	W. S. Harley	W. S. Harley
A. K. Brumbaugh	F. A. Whitten	A. J. Scaife	A. J. Scaife	A. J. Scaife	B. B. Bachman	F. A. Whitten
No Div. Apptd.	H. L. Pope	H. L. Pope	H. L. Pope	H. L. Pope	Discontinued	
J. J. Aull	Chas. Pack	Zay Jeffries	W. R. Webster	W. R. Webster	W. H. Bassett	Zay Jeffries
Clarence Carson	F. G. Whittington	W. C. Keys	W. C. Keys	W. C. Keys	H. S. Jandus	H. S. Jandus
Ralph Murphy	R. S. Begg	R. S. Begg	R. S. Begg	R. S. Begg	L. A. Chaminade	L. A. Chaminade
E. G. Budd	G. E. Goddard	A. J. Neerken	G. J. Mercer	G. J. Mercer	J. B. Judkins	Discontinued
					W. G. Careins	E. N. Sawyer
J. D. Harris	J. D. Harris	J. D. Harris	J. D. Harris	J. D. Harris	Consolidated with Engine Div.	
E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman
Consolidated with Parts and Fittings						
R. A. Schaaf	S. P. Hess	S. P. Hess	F. A. Whitten	F. A. Whitten	Discontinued	
T. C. Menges	T. C. Menges	L. F. Burger	L. F. Burger	L. F. Burger	Consolidated with Engine Division	
Bruce Ford	W. E. Holland	I. M. Noble	W. E. Holland	W. E. Holland	C. T. Klug	Consolidated with Elec. Equip.
J. G. Vincent	No Div. Apptd.	J. G. Vincent	J. G. Vincent	H. M. Crane	H. M. Crane	H. M. Crane
A. W. Copland	A. C. Bryan	A. C. Bryan	L. C. Fuller	L. C. Fuller	S. O. White	S. O. White

and Successive Chairmen of the Divisions

25	1926	1927	1928	1929	1930
ston	F. A. Whitten	K. L. Herrmann	H. M. Crane	A. J. Scaife	A. Boor
ly on	K. L. Herrmann R. S. Begg	C. M. Manly A J. Scaife	A. J. Scaife W. R. Strickland	G. L. McCain A. Boor	G. L. McCain C. A. Michel
titz	O. W. Sjogren	G. A. Young	G. A. Young	O. B. Zimmerman	Discontinued
ne	E. P. Warner	E. P. Warner	E. P. Warner	E. P. Warner	J. F. Hardecker
				L. M. Woolson	Arthur Nutt
on	C. C. Carlton	L. R. Buckendale	L. R. Buckendale	O. A. Parker	O. A. Parker
ey	H. E. Brunner	H. E. Brunner	H. E. Brunner	H. E. Brunner	G. R. Bott
e	H. S. Pierce	Discontinued			
					C. B. Jahnke
	T. L. Lee	B. M. Leece	A. R. Lewellen	A. R. Lewellen	D. M. Pierson
ll	C. R. Skinner, Jr.	C. R. Skinner, Jr.	C. R. Skinner, Jr.	Discontinued	
nan	Discontinued				
e	A. F. Milbrath	A. F. Milbrath	A. F. Milbrath	E. S. Marks	E. S. Marks
on	J. M. Watson	J. M. Watson	J. M. Watson	J. M. Watson	J. M. Watson
s	C. B. Jahnke	C. B. Jahnke	C. B. Jahnke	Discontinued	
el	C. A. Michel	C. A. Michel	C. A. Michel	C. A. Michel	C. A. Michel
gey	H. C. Mougey	W. S. James	E. W. Upham	E. W. Upham	E. W. Upham
on	C. A. Carlson	C. A. Carlson	L. Ochtman, Jr.	L. Ochtman, Jr.	L. Ochtman, Jr.
	A. J. Scaife	A. J. Scaife	A. J. Scaife	A. J. Scaife	Consolidated with Motor Truck Div.
					A. W. Herrington
rrington	W. S. Harley	W. S. Harley	W. S. Harley	Discontinued	
	B. B. Bachman	F. A. Whitten	F. A. Whitten	F. A. Whitten	Consolidated with Motorcoach Div.
	Discontinued				
ster	W. H. Bassett	Zay Jeffries	Zay Jeffries	Zay Jeffries	Zay Jeffries
	H. S. Jandus	H. S. Jandus	H. S. Jandus	A. Boor	A. Boor
	L. A. Chaminade	L. A. Chaminade	G. L. McCain	G. L. McCain	G. L. McCain
	J. B. Judkins	Discontinued			
	W. G. Careins	E. N. Sawyer	LeRoy F. Maurer	F. W. Stein	F. W. Stein
	Consolidated with Engine Div.				
n	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman	E. H. Ehrman
n	Discontinued				
	Consolidated with Engine Division				
d	C. T. Klug	Consolidated with Elec. Equip.			
	H. M. Crane	H. M. Crane	H. M. Crane	H. M. Crane	H. M. Crane
	S. O. White	S. O. White	S. O. White	P. L. Tenney	Consolidated with Parts and Fittings

SUCCESSIVE CHAIRMEN OF THE S.A.E. STANDARDS COMMITTEE

(1) Henry Souther, 1905 to 1914 (Deceased). (2) K. W. Zimmerschied, 1915. (3) A. Ludlow Clayden, 1916. (4) J. G. Utz, 1917 (Deceased). (5) B. B. Bachman, 1918 to 1921. (6) E. A. Johnson, 1922 to 1925. (7) F. A. Whitten, 1926. (8) K. L. Herrmann, 1927. (9) H. M. Crane, 1928. (10) A. J. Scaife, 1929. (11) A. Boor, 1930.

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that are of more or less direct interest to the Society's members and the automotive industry.

With the development of national standardization in most of the industrial countries of the world, it became apparent that international cooperation in standardization was desirable. Methods of correlating national standardization were discussed and an international group known as the International Standards Association was created in 1926, but the American Standards Association did not become a member of the I.S.A. until November, 1929. The I.S.A. is not an international standardizing body but is restricted to exchanging information between national standardizing bodies and acting as the agency for promoting international agreement on national standards so far as is practicable for the countries concerned.

Value of Standardization

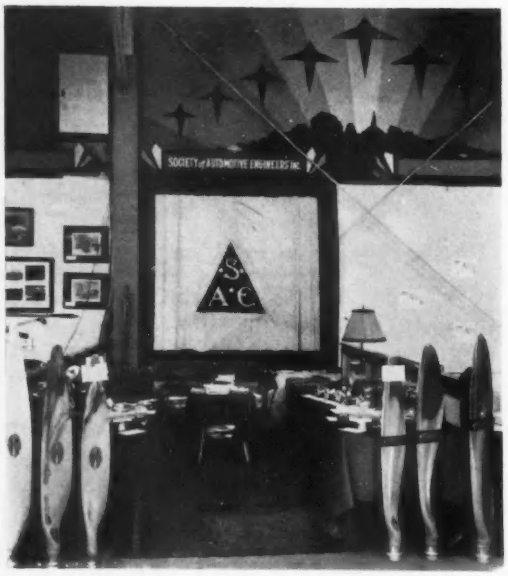
The benefits of standardization begin with the first operation of the fabrication of material or parts and increase in amount through the various stages of trade and commerce until the user of automotive apparatus is reached. He reaps the greatest benefit of all by being able to buy a better product at less cost and is taxed with greatly reduced maintenance expense.

The assignment of a definite value to the savings effected by standardization in the automotive industry would be a gargantuan undertaking, as it would necessitate a careful survey and analysis of the cost records of the entire industry over many years. However, an analysis of estimates made by nearly 150 engineers and executives of leading companies in the automotive field in 1921 indicated their opinion that the total annual retail cost of automotive products is at least 15 per cent less than it would have been without standardization. Applying this estimate to the annual wholesale value of complete motor-vehicles and parts and accessories, as compiled by the National Automobile Chamber of Commerce, the values would have been increased each year by the amount shown in the accompanying chart and the cost of the products to the consumer would undoubtedly have been proportionately larger each year.

The estimated cost of the standardization work during 1921 was $16,000 to the companies represented on the S.A.E. Standards Committee and $39,000 to the Society. The estimated cost for 1928 was $15,000 to the companies participating and $33,800 to the Society. Efforts have been made on a number of occasions to secure actual cost and savings records directly from the manufacturers, but little definite information has been available for even specific standards. The fact that standardization has been so well supported by industry for so many years shows, however, that this work has had a very real value and that it has increased from year to year.

Engineering Standardization Tomorrow

Standardization and the spirit of cooperation engendered by it have helped the automobile industry to make startling progress in a few short years such as has never been known before in industry. Standardization is progressing in the same way. A few years ago the industry was its chief exponent; today nearly all mechanical industries are realizing its great potentialities. Then it was largely a matter of company policy; today it is vital to the company, the industry, and the Nation—inter-industrially, nationally and internationally. Then it was the business of a few small committees; today its organization parallels that of colossal industries. Then it was incidental in the engineers' daily work; today it is becoming a National movement; and tomorrow it will be a power in industry and commerce that will be controlled and guided by master engineers and executives trained in the art of standardization.



25TH ANNIVERSARY

National Meetings of the Society

Facts and Figures Relating to Occurrences on the Road from Long Ago to Now

WHETHER successful meetings cause a society to flourish or a flourishing society produces successful meetings is a question of cause and effect as indeterminate as the well-known problem about the parentage of the egg and the chicken. Undoubtedly, in the case of a society and its meetings, it is less a matter of action and reaction than of interaction. In any event, the character of meetings arranged by an organization is indicative of the condition of that organization, and a study of the meetings staged by the Society points to a lively initial interest, a steady growth, a subsequent broadening and deepening of scope, and a healthy rounding out of activities to appeal to the entire automotive industry and the allied fields.

Founded 25 years ago, the Society of Automobile Engineers, as it was then called, held its first Annual Meeting in January, 1906, in New York City. The meeting consisted of one session at which three papers were read before an audience of 32. Parenthetically, it may be remarked that 1233 people attended the 1930 Annual Meeting, a five-day gathering at which 16 sessions were held and 29 papers were presented. The year 1907 saw a doubling in the number of National Meetings of the Society, because, in addition to the Annual Meeting held in New York City in January of that year, a Semi-Annual Meeting was held in July of the same year at Buffalo. The Semi-Annual Meeting, known as the Summer Meeting, immediately became an important feature of the Society's work, and each year since 1907 has witnessed a Semi-Annual as well as an Annual Meeting of the Society.

Initiation of Specialized Meetings

In 1916 a new development in Society meetings was initiated. Previous to that time, no National meetings other than the Annual and Semi-Annual Meetings were held, and they were then, as now, general in character, showing a cross-section of Society activity and not confined to any one specific interest. A Tractor Meeting was held in 1916, and since that time it has become a not uncommon occurrence for a National meeting to be devoted to some one phase of automotive activity. Thus, in 1917, the first National Aeronautic Meeting was held, and in subsequent years there have been meetings devoted respectively to such topics as motorboats, motorcoaches and motor-trucks, transportation and maintenance, and production.

Annual and Summer Meetings

The Annual Meeting was invariably held in New York City up to and including that in 1923, after which it was decided that Detroit should be the site of this yearly event. When the Annual Meeting was held in New York City, a dinner was an important feature of it and became the chief S.A.E. social gathering of the year. When the Annual Meeting was transferred to Detroit, attention was called to the fact that when the Society of Automobile Engineers became the Society of Automotive Engineers, Inc., a clause in the Constitution required that the Annual Meeting be held in New York. It was therefore decided that the Annual Dinner should continue to be held in New York City, and that it should convene at the opening session of the Annual Meeting and after a short business meeting should adjourn until a slightly later date in Detroit. Accordingly, the Annual Dinner is held during New York Automobile Show Week, and has become one of the outstanding social events of that busy week, being recognized in the industry as one of the largest and most representative gatherings of automotive engineers and executives held during the year.

Where to hold the Semi-Annual, or Summer, Meeting is a question of constant interest and ever-increasing difficulty that had to be considered and answered each year. Certain requirements must be met, and one at least of these requirements becomes more difficult to satisfy in direct ratio to the numerical growth of the Society membership. The location should be accessible to a main portion of the membership of the Society; the place chosen should have adequate accommodations for technical sessions; because of the semi-social character of the meetings, suitable recreational facilities should be available; and, because of the blending of the professional with the recreational features and the desirability of professional and social contacts during the meeting, the entire attendance should preferably be housed under one roof. This last-mentioned consideration excludes a number of places that otherwise would constitute excellent Summer Meeting sites.

Summer Meetings have been held at the following places: Buffalo, Detroit, Chicago, Dayton, O.; a boat trip to Mackinac Island, Mich., on board the City of Detroit II; a boat trip to Sault Ste. Marie, Mich., on board the City of Detroit III; Cape May, N. J.; a boat trip through Georgian Bay on board the Noronic; a boat trip to Mackinac Island and Killarney, Mich., on board the Noronic; the City of Washington; Ottawa Beach, Ill.; West Baden, Ind.; White Sulphur Springs, W. Va.; Spring Lake, N. J.; French Lick Springs, Ind.; Quebec, Canada, and Saranac Inn, N. Y.

Valuable as have been the general meetings of the Society to all the members and in fact to the entire industry, no less interesting and significant have been the topical meetings designed to appeal to some special portion of the membership.

Six Tractor Meetings Held

The first Tractor Meeting of the Society, which was held during a tractor demonstration at Fremont, Neb., August 9 to 12, 1916, was a joint meeting with the Society of Tractor Engineers. The meeting was convened to consider the advisability of the tractor engineers uniting in the proposed Society of Automotive Engineers. The consensus of opinion among the tractor men who attended the meeting was distinctly favorable toward the consolidation and the proposed merger was effected in due course. Subsequently, Tractor Meetings of National Scope have been held by the Society at Fremont, Neb.; Kansas City, Mo.; Columbus, O.; Minneapolis, and Chicago.

First S.A.E. Aeronautic Meeting

The first National Aeronautic Meeting, held by the Society in 1917, was prepared for by the consolidation in 1916 of the American Society of Aero-

nautic Engineers with the S.A.E. In connection with this important merger, it is of interest to read the following expression from the Board of Directors of the American Society of Aeronautic Engineers:

The broad-minded far-seeing manner in which the Society of Automobile Engineers has considered and acted on the proposal to form a general alliance of the organizations interested in fields concerned with the development and utilization of the internal-combustion engine is emphatically indicative of the new American spirit.

This proposed concentration of engineering effort upon the problems of the aircraft art cannot fail to insure the most energetic development of its important and rapidly growing industry.

That an organization which has already done so much for the Country and, in fact, the world, as well as for its own members, should take steps voluntarily to change its name and share its prestige with comparatively new organizations is a remarkable exhibition of idealism and deserving of broader fields of usefulness and still greater achievement.

The Society of Aeronautic Engineers desires to express its admiration and appreciation and, in addition, its hope that it may be able to do its share in the broader movement which, through your cooperation, is now under way.

PART OF GROUP OF MEMBERS GATHERED AT THE MARQUETTE STATUE ON MACKINAC ISLAND DURING THE SUMMER MEETING CRUISE IN 1916

Previous to the National Aeronautic Meeting in 1917, an aeronautic paper had been presented before the Metropolitan Section and another paper on an important phase of aeronautic work had been delivered at the 1916 Summer Meeting; also, the Aeronautic-Engine Division of the Standards Committee had been formed and held its first meeting on June 18, 1916. With aeronautic activity thus becoming an important part of the Society's work, it was fitting that increasing attention should be given to the scheduling of aeronautic papers at Annual and Semi-Annual Meetings and to the holding of National Aeronautic Meetings. Accordingly, aeronautic topics have occupied a suitably prominent place on the programs of these meetings, and Aeronautic Meetings of National scope have been held as follows:

National Aeronautic Meetings

Date	*Location*
1917	New York City
1919	New York City
1920	New York City
1924	Dayton, Ohio
1925	New York City
1926	Philadelphia
1927	Spokane, Wash.
1927	New York City
1928	Los Angeles
1928	Chicago
1929	Detroit
1929	Cleveland
1930	Miami, Fla.
1930	St. Louis
1930	Detroit
1930	New York City
1930 (scheduled for August)	Chicago

Value of Production Meetings

The holding of National Production Meetings was begun in 1922, when the Society staged a Production Meeting in Detroit in October. The Meetings Bulletin containing an announcement of the first Production Meeting expressed the idea that prompted the inauguration of this practice of holding Production Meetings annually and that has motivated its continuance. A pertinent excerpt from the announcement is quoted herewith:

Devoted exclusively to papers and visits of interest and value to the production man, it is expected to attract factory managers, superintendents, efficiency men, tool builders and production engineers from the entire automotive industry. . . . The administration, sales, service and engineering branches of the industry have long appreciated the advantages of meetings of this sort. Nothing is more beneficial than talking over your troubles and experiences with other men engaged in the same labors as yourself. The war showed us the value of cooperative exchange of production knowledge. The Annual S.A.E. Production Meeting will provide the opportunity for that same swapping of ideas, shop kinks and experiences which was so abruptly removed with the end of the war.

Production Meetings have been held as follows:

Production Meetings

Date	*Location*
1922	Detroit
1923	Cleveland
1924	Detroit
1925	Cleveland
1926	Chicago
1927	Cleveland and Detroit
1928	Detroit
1929	Cleveland
1930 (scheduled for October)	Detroit

Heavy-duty vehicles have received due attention at National Meetings. Besides papers and discussions at Annual and Summer Meetings, meetings of National scope have been devoted to problems con-

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SEVERAL OF THE NOTABLE GATHERINGS IN THE HISTORY OF THE SOCIETY

(1) ATTENDANCE AT A MEETING OF THE SOCIETY 22 YEARS AGO

This Meeting Was Held in Detroit in June, 1908, Three Years After the Organization of the Society. Among Those in the Group Are Henry F. Leland, Thomas J. Fay, Charles T. Hayward, E. S. Foljambe, F. J. Newman, H. W. Alden, C. D. Cramp, B. G. Ellis, C. A. Trask, William G. Wall, Bruce Ford, E. E. Sweet, G. E. Franquist, Frank Johnson and H. K. Holsman

(4) AIRPLANE PHOTOGRAPH OF MEMBERS AT THE 1926 SUMMER MEETING AT FRENCH LICK

The Members Formed into the Initials of the Society on the Field in Front of the Hotel and the Photograph Was Exposed, Developed and Printed in 15 Min. in an Airplane from McCook Field, Dayton, and Dropped from a Height of 1000 Ft. to within a Few Feet of the Flag Man Waiting To Receive It. The Picture Was Shown on the Screen at the Evening Session the Same Night

(2) BOARDING THE CITY OF DETROIT III FOR THE 1913 SUMMER MEETING AND CRUISE ON THE GREAT LAKES

(3) S.A.E. MEMBERS GATHERED ON THE BEACH AT THE SUMMER MEETING AT SPRING LAKE, N. J., IN 1923

(5) THE INTER-SECTION CHASSIS-ASSEMBLING CONTEST WHICH WAS ONE OF THE OUTDOOR EVENTS AT THE SUMMER MEETING AT FRENCH LICK IN 1927

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nected with mass transportation of goods and people by motor-truck and motorcoach. Sometimes design has been the topic discussed, sometimes operation, sometimes maintenance, and sometimes a combination of these topics. Two Truck Meetings were held in 1919, one in Chicago and one in New York City, and a Truck and Tractor Meeting took place in Chicago in 1920.

Service and Transportation Meetings

Service Meetings were held in Chicago in 1922 and 1923, in Dayton in 1923, in Detroit and Cleveland in 1924, and in Chicago in 1925. The holding of meetings devoted exclusively to service was discontinued in 1926, with the idea that this phase of automotive engineering should be included in the Automotive Transportation Meeting, the annual holding of which had been inaugurated in 1923. As implied in the previous sentence, the Automotive Transportation Meeting was originally designed for a discussion of operation topics, but beginning with 1926 it has discussed both operation and maintenance. Automotive Transportation Meetings have been held at follows: 1923, Cleveland; 1924, New York City; 1925, Philadelphia; 1926, Boston; 1927, Chicago; 1928, Newark; 1929, Toronto.

Impetus was given to the transportation and maintenance meetings work by a Constitutional amendment passed in 1925 which provided that engineers chiefly concerned with the utilization of automotive apparatus should be eligible to membership in the Society equally with the designers of such equipment. The way had been prepared for this recognition of the importance of the operation man in automotive engineering by the trend of thought expressed in the Presidential address of the late Charles M. Manly, in 1919. In that address Mr. Manly spoke in part as follows:

> But the work of the automotive engineer is not finished, but merely begun, when the machines are designed and built, whether they be motor-trucks, tractors, aircraft or other vehicles. There is more need today for real engineering work in connection with the planning and organization of the operating end of automotive vehicles and machines than there is in connection with the design and construction of them, just as there are more engineers engaged in the operating and maintenance end of railroads than are engaged in the design and construction of locomotives and railroad cars.
>
> In the subdivision of automotive engineering work having to do with motor-trucks, the real work of the engineer has hardly as yet been begun. True it is that motor-trucks are being sold and are daily hauling thousands of tons of merchandise and general freight, but the careful study and collection of data for accurately predetermining the best operating equipment, organization and personnel to meet given conditions at a definitely predetermined cost has hardly been started. This single phase of automotive engineering presents more problems for the engineer to solve them would be needed if all our records and data in railroad transportation engineering were suddenly swept away and it became necessary to re-establish such data immediately for the determination of proper freight rates.

National Motorboat Meetings

The first National Motorboat Meeting of the Society was held in New York City in January, 1918. Since that date, Motorboat Meetings have been held by the Society in New York City in February and December, 1920, in 1922, 1923, 1924, 1925, 1929, 1930, and in Detroit in 1929.

The Meetings Committee

Time and thought have always been given unstintingly to the Society's events by the members of the Meetings Committee, and the success of the meetings is in no small measure due to this fact. The function of the Meetings Committee has been to arrange the program of each National meeting and to have general charge of the entertainments provided for the members and guests. In arranging the programs, it has been the duty of the Committee to procure professional papers, pass upon their suitability for presentation and suggest topical subjects for discussion at the meetings.

When the divisional reorganization of the Society was effected in 1929, the work of formulating the technical programs of National Professional-Activities Meetings and of special professional-activity sessions of Annual or Semi-Annual Meetings was transferred to the Professional Activities Committees. It is the duty of the Meetings Committee to pass upon all matters of policy pertaining to the meetings of the Society, arranging the general program and, except as just mentioned, formulating the technical programs of the Annual and the Semi-Annual Meetings, with general charge of the entertainment to be provided. The following members have served as Chairmen of the Meetings Committee: A. L. McMurtry, H. M. Swetland, E. T. Birdsall, A. B. Cumner, Robert McA. Lloyd, David Beecroft, C. F. Scott, M. P. Rumney, T. J. Lítle, Jr., L. Clayton Hill and John A. C. Warner.

Meetings Keep Pace with Industry

Recognized as an important factor in the life of any really active organization, meetings are particularly valuable in furthering the interests of engineers engaged in the automotive industry because of the rapidity of its progress. What is new and uptodate at one time may be superseded in the proverbial twinkling of an eye, and a

VISITING DELEGATION OF INSTITUTION OF AUTOMOBILE ENGINEERS OF ENGLAND AND MEMBERS OF

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wide-awake engineer is eager to take advantage of the opportunity afforded him by S.A.E. meetings to maintain an intimate contact with the rapid advances that are constantly being made. Viewed in this light, attendance at S.A.E. meetings is rightly looked on, not as an obligation that the member owes to the Society, but as a privilege with tangible benefits that he does not willingly deny himself.

Technical programs at S.A.E. meetings have served as an authoritative medium for the first detailed information concerning a great variety of new developments. To cite one example, members who attended the 1923 Summer Meeting at Spring Lake, N. J., will recall the papers on balloon tires and on four-wheel brakes, together with the demonstrations of these departures from the established order. Now that balloon tires and four-wheel brakes have become as much a matter of course as the automobile itself, it is interesting to recall that prior to the 1923 Summer Meeting the general public knew virtually nothing about these two developments, and automotive engineers themselves knew very little more about them. A later example is concerned with the recent developments in Diesel engines, particularly the development of the Diesel engine for aircraft. In fact, meetings of the Society have been of unusual value in the introduction of information that definitely exemplifies the rapid progress in the field of aviation. For instance, the all-metal airship developed for the United States Navy was brought to the attention of engineers and others through S.A.E. meetings.

Progress Reports on Research

One of the greatest pieces of work in which the Society has ever engaged is the fuel-research program, carried on in cooperation with the National Automobile Chamber of Commerce, the American Petroleum Institute and the Bureau of Standards. This important study has had for its guiding principle the mutual adaptation of the fuel to the engine and the engine to the fuel, to the end of National economy and efficiency. In a paper read at the 1929 Summer Meeting, Dr. H. C. Dickinson of the Bureau of Standards, reviewed briefly the causes leading up to the research, summarized the progress made in the several phases of the work and mentioned briefly the plans for continuation of the program. Dr. Dickinson emphasized that this research work had brought into accord the engineering staffs of the automotive and petroleum industries, contributed largely to fuel saving for the Country, developed the data needed to combat crankcase-oil dilution, provided complete data on the relationship of fuels to engine-starting and cleared up the question of the effect of fuel on engine acceleration. An important problem now receiving study is concerned with securing satisfactory control over the relationship between fuels and compression ratios.

Results of this research program have been made known to the membership of the Society through the medium of papers and progress reports delivered at its National meetings. The first of the papers was presented at the 1923 Annual Meeting, and every Annual and Summer Meeting program since that date has carried one or more of these valuable papers and reports. The far-reaching significance of these investigations and the superlative importance of the reporting of the results through the meetings can be appreciated only if one stops to consider what might have happened to the fuel situation if present-day engines had not been capable of handling the cracked fuels developed to conserve the Country's fuel resources.

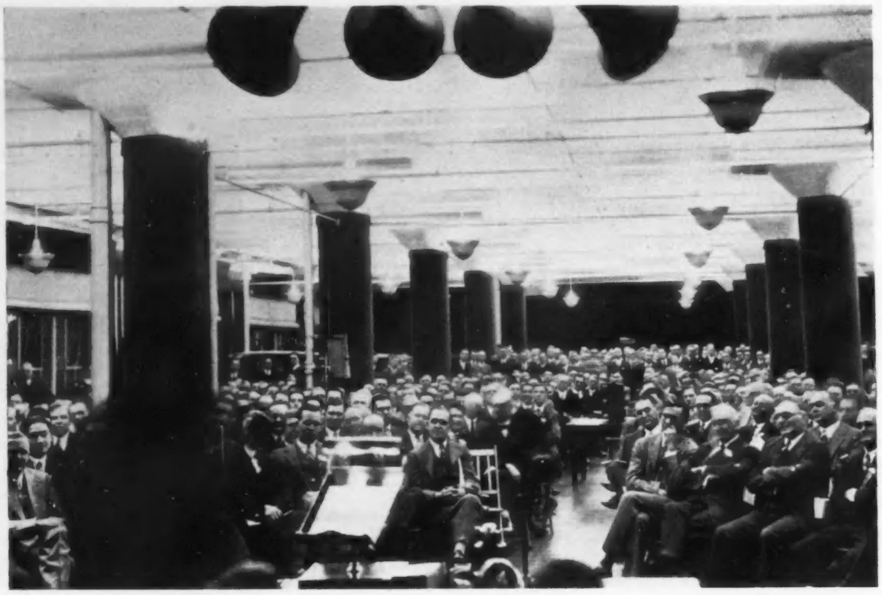
A great many excellent papers on subjects less fundamental but none the less valuable in their fields have been brought to the notice of the members and the industry through meetings of the Society. Members who make it a practice to attend National meetings of the Society will recall exceedingly informative papers that have been given at one time or another on such varied yet pertinent topics as valves and valve design, as well as papers on topics belonging to the more specialized fields such as aeronautics, tractors, marine engines, motor-trucks and motorcoaches, automobile bodies and the like.

Relations with Foreign Engineers

Foreign developments in automotive work have been brought before the engineers in this Country through the medium of programs at S.A.E. meetings. To mention a few specific examples, the members who attended the 1926 Aeronautic Meetings listened to a paper on economical production of airplanes and seaplanes that had been prepared by Adolph Rohrbach; those who attended the 1928 Annual Meeting heard Dr. Wilhelm Riehm's paper on high-speed automotive Diesel engines and D. Sensaud de Lavaud's paper on independent springing, steering and shimmy. A paper by M. de Lavaud on his automatic transmission was presented at the 1928 Summer Meeting. Harry Ricardo spoke at the 1922 Annual Meeting; Andre Citroen visited this country and spoke before S.A.E. members in 1923, as did Alan Fenn and Sir Alan Cobham in 1927, to cite but a few such cases.

At this writing the Society is anticipating a visit from 30 French engineers, members of the Société des

THE S.A.E. ON AN INSPECTION TRIP TO THE CHALMERS FACTORY IN DETROIT ON JUNE 4, 1913



AN INTERESTING AND INSTRUCTIVE FEATURE OF THE 1929 ANNUAL MEETING WAS A VISIT TO THE CHRYSLER LABORATORIES IN DETROIT AND A RESEARCH SESSION HELD THERE

Ingénieurs Automobiles, to attend the Summer Meeting at French Lick, May 25 to 29. Although no paper by any of these men was scheduled, it is believed that the establishment and confirmation of very cordial relations between the Society and its sister organization in France will result.

This visit of the French engineers will serve to recall to many the trip to Europe made by members of the Society in 1911. A contemporary account of the visit gives the following information:

> The first stop was made in England, and then the journey was continued to France. The reception accorded the Society members was most cordial and very elaborate. They were admitted to the most famous factories that never before had been opened to outsiders. This visit was a very profitable one, for it resulted in the acquisition of much technical information, and many valuable acquaintanceships were made between the members of the Society and the heads of the factories visited. This was the opening wedge on the other side of the Atlantic to securing an interchange of information and ideas between American and European engineers. Naturally, the result of this exchange of ideas benefited both American and European industries far more than was appreciated at that time.

As Much to See as to Hear

In addition to disseminating uptodate information through the medium of stimulating and informative papers, National meetings of the Society have presented opportunities to the members to see instructive exhibits and demonstrations and to go on interesting and thought-proving inspection trips. "As much to see as to hear" has been the slogan, expressed or implied, of many of the National meetings, and the exhibits and demonstrations found at the meetings have proved the slogan to be more than a matter of mere words.

So many, so varied and so useful have been the inspection trips arranged by the Society for the benefit of members attending the National meetings that it would be difficult to enumerate all of them and futile to select a few for special mention. The value of such a feature on a program was brought out in an editorial item published in the May, 1930, issue of the S.A.E. JOURNAL, p. 542, from which the following excerpt is made:

> Inspection trips through large modern industrial plants are both instructive and interesting and lend special zest to technical papers presented by engineers of plants visited. They also serve to get the members out of the particular rut of their individual work, giving them new ideas and stimulating their enthusiasm in their jobs.

Intangible Benefits from Meetings

The value of meetings has been expressed in the foregoing in terms of papers, demonstrations, exhibits and inspection trips, but such a summing up of the importance of technical gatherings falls short of telling the whole story. These are indeed the more obvious features, the items that can be checked on a printed program or read later in a report of the meeting. But other factors, equally or more important though often less tangible, enter into the success of a meeting, its value to those who attend and the consequent influence on the industry. In the opinion of the members whose interest in Society meetings is more than casual, these less readily measured factors constitute the chief advantage to be derived. Incidentally, it may be said that this advantage must be gained by actual attendance at a meeting and in no other way, no matter how interesting and accurate a published report of a meeting may be.

A little reflection shows that one success-making factor of outstanding value is related to the quantity and the quality of discussion following the presentation of a paper or a group of papers. Organized, prepared discussion, which is announced in the printed program, is a recognized part of a technical session, and an appreciation of its value was included, by implication at least, in the foregoing summary of the importance of papers presented at meetings. But many members feel that they gain more than can be estimated from the impromptu and informal personal discussions which take place before, during and after a meeting.

As has been said more than once, it is "the session after the session" that really counts. The reason is not hard to find. Engineers are for the most part controlled by the business policies of their respective companies and for that reason hesitate in many instances to say what they think, either before a large audience or in print. But when they gather in small groups informally, before and after the session, the facts are likely to come out. Participating in such impromptu and informal personal discussions, whether as a listener or as one of the talkers, brings a benefit to the participant that he could not have gained if he stayed away from the meeting with the mistaken idea that he could get all the value out of the meeting by reading the papers and the printed discussion afterwards.

Contacts Stimulate Independent Thinking

Another important advantage to be derived from attendance at a meeting is that the association with men of similar interests, as well as the listening to papers and discussions, puts an engineer into a current of thought which stirs his imagination and makes it easy for him to do some independent thinking that may result in creative enterprise which is likely to redound to his credit and the profit of his company.

Meetings, moreover, offer an engineer an opportunity for self-expression. It has been said that practical men are too often in the class of those "unaccustomed to public speaking" and that far too often they actually seem to pride themselves on this fact. Yet

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the progressive engineer, albeit a practical man, is a man concerned with ideas, their conception and their promotion. He may develop an idea in solitude and silence, but when he starts to promote it or, in common parlance, "put it across," he finds himself obliged to describe, narrate, expound and, frequently, argue. Many an engineer has found that meetings of the Society have given him an opportunity to gain experience in presenting concisely and adequately what he has in mind. The mere taking part in the discussion at a technical session, speaking before a group of men interested in the same line of work, outlining what he thinks or believes or knows, is a genuine help to an engineer, giving him practice in formulating his ideas and developing them to a logical conclusion.

It is a truism to say that a meeting should, and usually does, give people a chance to meet, and, like most facts hidden behind truisms, its significance is in danger of being overlooked. Many members are frank to admit that one of the chief points of value in Society work is the acquaintanceship that social contact at the meetings develops. Such contact enables an engineer to meet his competitor on a friendly basis, with a consequent smoothing out of business relationships. One of the chief causes of misunderstanding among human beings, as has been pointed out, is their failure to get together and discuss their common problems on friendly terms and in a legitimate way. An opportunity for such gathering and such discussing is offered to automotive engineers, both individually and collectively, through the medium of Society meetings.

Throughout the 25 years of the Society's life, many members have taken advantage of the opportunity offered by the meetings to discuss their problems, sometimes giving help and sometimes receiving it. The results of experiment and research have in countless cases been contributed for the general good, and the inspiring progress of the automotive industry is an example of the beneficial effects of community thinking. The industry believes in the liberal interchange of information and ideas, and the meetings of the Society have helped to build up that creed and in turn have felt its influence.

SOME OF THE SECTION BOOTHS AT THE COUNTRY FAIR HELD IN THE CHATEAU FRONTENAC, QUEBEC, AT THE SUMMER MEETING IN 1928

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Review of S.A.E. Research Activities

Initiation, Progress and Results of Programs That Have Kept Pace with Advancement of the Industry

ALTHOUGH the United States has shown by far the greatest industrial development in the automotive field, this Country contributed very little to the sum of automotive research during the early years of the industry. The engine and other design features were more a result of experimental development than of fundamental research. This situation is not difficult to explain. The automobile was very favorably received and the manufacturers were kept busy producing to meet the demand. There was no need for improving design in the direction of greater economy or efficiency. In Europe there was keener competition in the details of design, and consequently our engineers relied upon the work of British, French and German investigators to meet American needs. The information available became inadequate, however, and there was a general awakening to the need, not only of research in the interest of practical and profitable results, but of pure research, the advancement of knowledge in a particular problem affecting the industry as a whole.

Research Division of Standards Committee

A recognition of this need on the part of members of the Society and the Council resulted in the formation of the Research Division of the Standards Committee in 1914. It was thought that, after all available data had been collected by the divisions of the Standards Committee on given subjects, some unsettled points could be determined by tests which it would be feasible to have made through the Research Division, the members of which occupied chairs at various institutions of learning. D. L. Gallup, professor of gas engineering at the Worcester Polytechnic Institute, was appointed first chairman of the new division. The first set of tests were undertaken to determine the proper sizes of tap drills, a subject which the Data Sheet Division took up with a view to decreasing the breakage of taps and which, after a preliminary investigation by the Data Sheet Division, was by its request referred to the Research Division. Progress reports were rendered on this and other subjects, such as the development of a taxation formula to be applied in connection with vehicles, both motor driven and horse drawn, and standard form for a complete car-performance test, including tests for fuel economy and acceleration.

The activities of this Division were dropped of necessity in 1917, when the members of the Committee became too busily engaged in war work to allow time for service on such other projects.

Studies Directed by Research Committee

When the Society was again free to turn its attention to research, a committee was formed entirely separate from the Standards group, to be known as the Research Committee. This Committee has, during the 11 years of its existence, directed investigations on a number of problems in several phases of automotive engineering. Foremost among these projects is the cooperative fuel research, a study of engine performance and fuel characteristics that has been carried on at the United States Bureau of Standards under the direction of a Steering Committee made up of representatives of both the automotive and petroleum industries.

The Committee has also cooperated with the United States Bureau of Public Roads in conducting motor-truck impact tests and has had technical direction of headlighting research conducted at the Bureau of Standards to determine the basic requirements for safe and satisfactory headlighting, with due regard to protection against glare.

In the field of riding-qualities, the Research Committee functioned for some time as a medium of exchange of information on investigations being made by its own members and others, and has recently undertaken an extensive program of research covering a number of very interesting phases of the problem.

The carbon-monoxide content of exhaust gases and its effect on atmospheric pollution, and the fundamentals affecting wheel alignment are also among the subjects of investigation under the jurisdiction of the Research Department and Research Committee.

As early as 1919 the demand for gasoline began to push the supply and the quality began to change. No reliable information was at hand at that time as to the effect of gasoline volatility on engine performance, and a general conference was called to formulate a definition of gasoline. With the intervention of the World War, the question of gasoline specifications was side-tracked, but in 1918 complaints concerning gasoline were louder than ever. There were claims and charges between the technical men in the automotive and petroleum industries, but no information was available as to how the fuel behaved in the engine or what either industry might properly expect or demand of the other.

Fuel Investigation Started

At this stage of the controversy, a session at the 1919 Annual Meeting of the Society was devoted to a consideration of the problem. As a result of the discussion, the Council established a Research Committee charged with considering ways and means for active work on the fuel problem. The Committee was headed by B. B. Bachman and held its first meeting in the City of Washington on April 11, 1919.

At the second meeting of the Committee, held in New York on April 24 of the same year, it was moved that a committee to be known as the Automotive Fuel Committee be formed to consider the fuel situation and promote research intended to aid in the problem of assuring an ample supply of engine fuel at a price favorable to the continued growth of automotive transportation. This was the birth of the Cooperative Fuel Research, which began and has continued as a four-sided undertaking. The National Automobile Chamber of Commerce and the American Petroleum Institute, representing the two industries, furnished the funds. The Society of Automotive Engineers undertook to administer the funds, and the Bureau of Standards consented to carry on most of the research work. A joint steering committee was appointed to confer on program and pass upon the results. The complete story of this undertaking is told by Dr. H. C. Dickinson in a paper, presented before the

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Society, The Cooperative Fuel Research and Its Results, at the 1929 Summer Meeting.

Some Fuel-Research Results

Dr. Dickinson points out that in the early conferences controversy between the two industries was the order of the day, but each meeting of the steering committee brought the two groups closer together, until harmony and cooperation among the engineers of the two industries has become an outstanding feature of the project.

Some of the tangible results of this research are: improved quality of gasoline, together with an increase in yield per barrel of crude; coordination of engine design and fuel characteristics, resulting in easier starting, better acceleration and improved performance in general; significant steps toward preventing crankcase oil dilution and its accompanying ills; and the prospect of writing gasoline specifications based on fundamental knowledge that will have a true bearing on car performance.

Undoubtedly the contact between the two groups has been largely responsible for the establishment by petroleum companies of laboratories equipped for effective study of the relationship of fuel characteristics and engine performance, whereas in 1922 such laboratories were almost unknown.

Subcommittee at Work on Detonation

In February, 1928, the Cooperative Fuel-Research Steering Committee organized a Subcommittee on Methods of Measuring Detonation, and on it was cast the burden of developing an apparatus and a method for making knock ratings that would be universally acceptable, together with the task of standardizing a scale of reference fuels. The hope was that this undertaking would result in a common basis for expressing knock-test results, which, under the present conditions of diversified apparatus, methods and reference standards, is altogether out of the question.

The first phase of this research is nearing completion; that is, the development of a suitable engine for use as a standard instrument in determining the knock rating of fuels. These experimental engines are now in operation in the laboratories of the members of the Subcommittee, and work during the last few months has resulted in the development of equipment and accessories to complete the set-up. A series of tests is being carried out using the representative knock-testing methods to determine the suitability of the engine for use with any of the methods now in common practice and to ascertain the effect of engine variables on knock rating.

For the first time in the history of detonation research, several laboratories are working with identical equipment and the same fuels, and are using the same technique. Furthermore, in response to an expression of interest and a suggestion of cooperation from a group working along similar lines in Great Britain, the Steering Committee extended an invitation to the British group to join the Detonation Subcommittee in carrying out the proposed tests. The acceptance of this group has been received, and arrangements are under way to supply the new members of the Committee with identical equipment and test fuels. This last step makes the project an international one and paves the way for the second phase of the program, which will involve determining the relative merits of the various methods of measuring knock and, if possible, establish some method or methods as standard procedure that will be universally acceptable.

The Vapor-Lock Problem

At present the vapor-pressure data secured in the course of the fuel research are being applied and extended to solve the problem of vapor lock. Under the direction of the Fuels-Research Subcommittee, research on vapor lock in airplane fuel-systems as affected by fuel characteristics was undertaken early in 1929 at the Bureau of Standards with funds provided by the Naturaline Co. of America. This project has now been taken over by the Cooperative Fuel-Research Steering Committee, and the investigation is being extended to cover the conditions that exist in automobile engines wherein the phenomenon of vapor lock is frequently encountered but does not present so great a hazard as when it occurs in the airplane powerplant. Progress reports on this work have been presented before the Society during the last year.

Diesel Fuel-Oil Research

At a recent meeting of the Fuels-Research Subcommittee, a program of Diesel Fuel-Oil Research was adopted and plans for conducting and financing this project are now being perfected.

Although the fuel problem was doubtless the most pressing and important one facing the automotive industry at the time the Research Committee was organized, the Committee was appointed to deal with research problems of a general nature as well as the fuel problem and, in response to a request from the Bureau of Standards, to act in an advisory capacity in the direction and conduct of research work in the automotive fields. In 1921 the Research Committee was designated as a separate and distinct group from the Fuel Committee, the fuel group being given the status of a research subcommittee. H. M. Crane was appointed Chairman of the reorganized Research Committee.

Research Department Organized

Also in 1921 the Research Department was organized to assist the Research Committee in the general advancement of research, and its establishment constitutes a step of the first order of importance in the history of the Society.

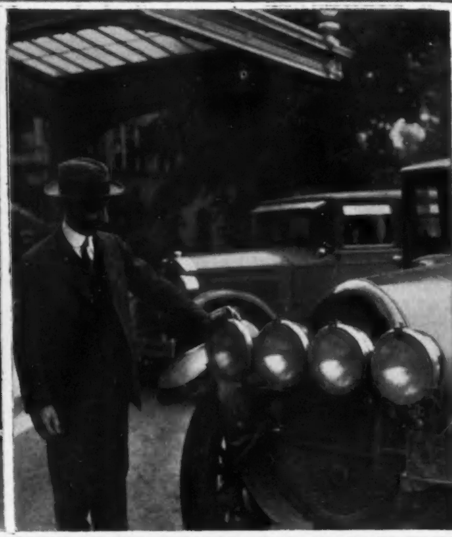
The principal bases of operations for pure research are the universities. A review of the research field made nine years ago revealed a number of organizations, both schools and manufacturing laboratories, engaged in interesting investigations of their own, without any relation to one another. Every day valuable results were being obtained and only a few individuals were getting the benefit of them. Manufacturers were looking vainly for information that was available but did not know where to go for it. Work was being over and over duplicated because those doing the work had no idea of what others were doing, while more important and pressing problems were awaiting solution. The Research Department of the S.A.E. was organized to serve as a clearing-house of information on research in progress at college and industrial laboratories and to make known the results through the Automotive Research columns of the S.A.E. JOURNAL, through papers presented at the Society meetings and through the work of the Research Department.

Dr. H. C. Dickinson relinquished service with the Government at the Bureau of Standards, to which he has now returned, so that he might organize and take charge of the Society's newly formed department.

The Research Department attends to the executive details connected with carrying out the plans of the Research Committee. In addition, through the Research Manager, contact is maintained with the research work carried on at universities, and articles on related research topics appear monthly in the Automotive Research section of THE JOURNAL.

In consequence of a lively and increasing demand for technical information, the Society established a technical information service. From its inception this branch has undergone a continued enlargement of its sphere of

APPARATUS USED IN HEAD-LAMP TESTS IN MARCH, 1918
These Tests Were Made Cooperatively by the Lighting Division of the S.A.E. Standards Committee, the Automobile Head-Lamp Committee of the Illuminating Engineering Society and the Electrical Testing Laboratories



DR. H. C. DICKINSON AND A TEST SET-UP THAT WAS USED IN NIGHT DEMONSTRATIONS AT THE 1927 SUMMER MEETING

activities as the demands made upon it have increased. During 1929, 1600 letters were written answering individual requests for information, and hundreds of telephone inquiries and visitors to the Society's library were taken care of. There queries may sometimes be disposed of by a simple statement of fact or formula or may require hours of searching or the preparation of a bibliography.

To facilitate this service, about 195 separate issues of technical magazines, including the leading foreign automotive journals, are perused each month and the articles of interest indexed for the card catalog that serves as a cumulative index to the S.A.E. JOURNAL, TRANSACTIONS, and a list of other important technical publications. From this material the best articles are selected and abstracted each month for the Notes and Reviews columns of THE JOURNAL. Books of technical interest are also reviewed and added to the library, and a selected number of magazines are bound and kept permanently in the Department.

During the course of the Department's work in coordination with the Research Committee, subjects of interest have been given special attention and extensive searches made of available literature relating thereto. Notable among these projects are brake testing gears, riding comfort, and carbon monoxide in exhaust gases.

S.A.E

In 1925 the Department materially assisted the work of the Cooperative Fuel-Research Steering Committee in its survey of actual conditions of crankcase-oil contamination existing throughout the Country. A group of carefully selected service stations cooperated in the collection of samples of contaminated crankcase-oil, all the stations following a simple uniform procedure in collecting the samples. The Bureau of Standards analyzed 656 samples and submitted the data to the Research Department, which carefully studied them and summarized the results in a paper presented at the 1926 Summer Meeting.

Highway Research Begun in 1921

In 1921 a score of National organizations were asked to appoint representatives on the Advisory Board on Highway Research of the National Research Council. At that time there was urgent need for a systematic and comprehensive research program to develop a sound scientific engineering basis for road building and to coordinate the efforts of the numerous existing agencies. The Society of Automotive Engineers has continued its representation on the Advisory Board up to the present time, a whole series of valuable reports on the subject having been published in the interim.

In the latter part of 1921 the Council of the Society designated H. W. Alden as chairman of a committee of five, to be known as the Highways Research Subcommittee, to represent the Society in cooperative work on highway matters with the Government and engineering and trade organizations, and to sponsor independent research.

About the same time Dr. T. J. MacDonald, chief of the Bureau of Public Roads, called a conference to consider a program of highway research laid out by the Bureau. This provided for a series of impact tests to determine the maximum stress to which different thicknesses of concrete road on different types of subgrade can safely be subjected, and what classes of vehicle, as regards weight, speed, class of tires and design, can be operated without exceeding this safe limit of stress.

The S.A.E. Highways Research Subcommittee has acted in an advisory capacity on the Cooperative Committee on Motor-Truck Impact Tests, together with the Rubber Association of America and the Bureau of Public Roads, in carrying out this program. Reports on the work have been published from time to time in the S.A.E. JOURNAL and other publications. During the last two years the Bureau of Standards has cooperated on problems of instrumentation connected with the tests, and a report covering the work to date is now in the final stages of preparation for publication.

Cooperative Headlighting Tests

Since adequate head-lamps undoubtedly enlarge the sphere of usefulness of motor-cars, the complex question of what is satisfactory light distribution for automobile driving has received a large share of attention from the industry. As early as 1914 the subject of glare was being discussed at S.A.E. meetings, and shortly afterward the Lighting Division of the Standards Committee concerned itself with the problem. In November, 1917, this division held a

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joint meeting with the Automobile Head-lamp Committee of the Illuminating Engineering Society, and this joint committee, with the cooperation of the Electrical Testing Laboratories, arranged for a road test with controllable head-lamps and photometric apparatus to record the actual intensity of light in foot-candles. The tests were held in March, 1918, on a short stretch of asphalt road between Pelham Parkway and the Morris Park station of the New York, Westchester & Boston Railroad. Invitations were extended to many of the Eastern State motor-vehicle officials, and the results were discussed at a meeting the following month.

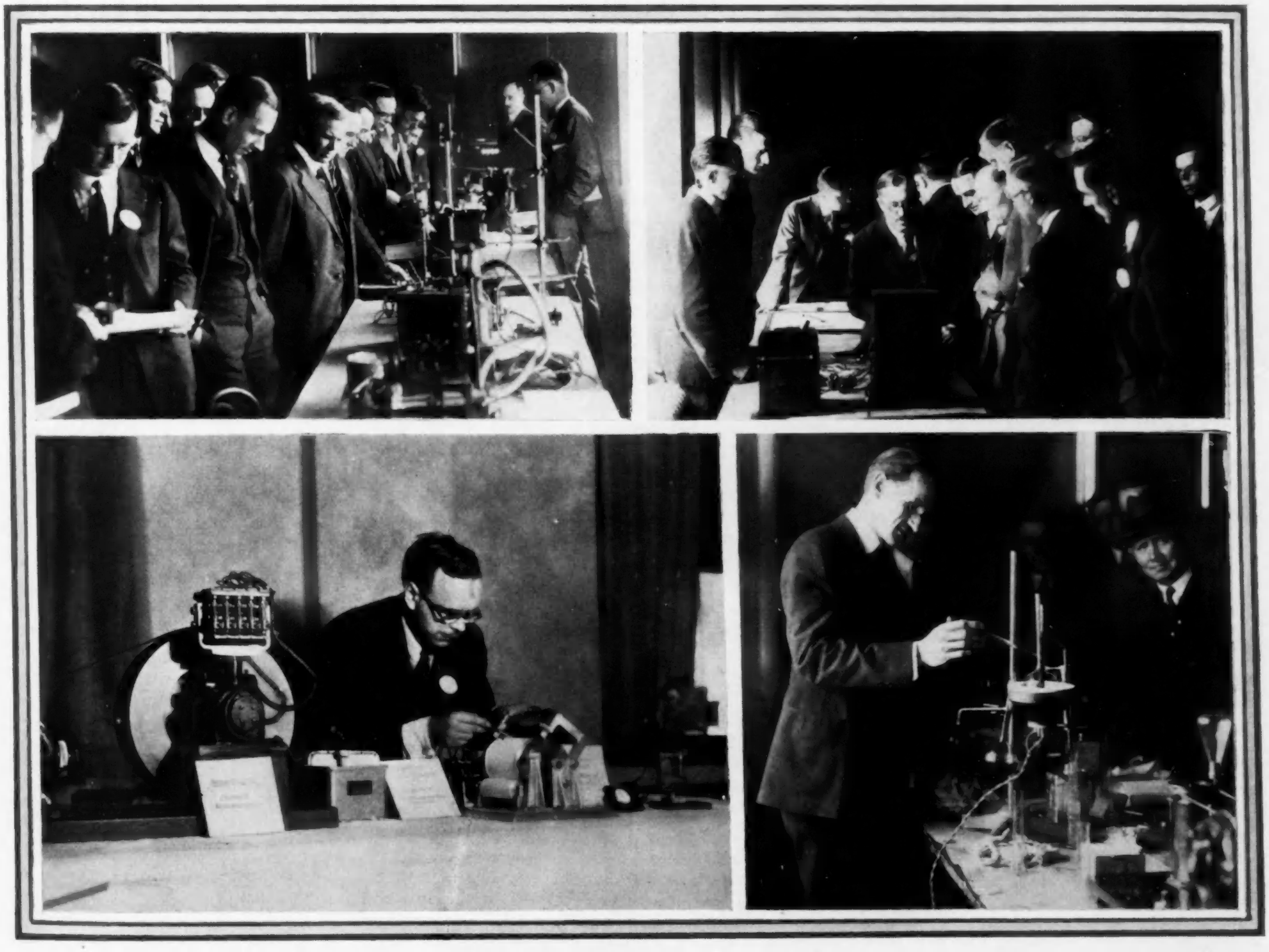
The object of the preliminary work was to determine two points; first, the minimum amount of illumination necessary, for the safety of both driver and pedestrian, to reveal from the driver's seat a man in dark clothes in the road 150 and 250 ft. in front of the car; secondly, the intensity of glare a driver can tolerate in his eyes and still see the portion of road beside the approaching car and over which he must drive in passing.

These tests were conducted with a view toward yielding information on which to base legislation governing headlight specifications which was then being enacted by some of the States. The Standards Committee made recommendations based on these data and continued its interest in the research side of the project until the Research Committee was formed and took over this phase of the problem.

In 1925 the Research Department began preliminary work on this subject, such as collecting data on headlighting regulations then in existence in the different States. The following year the Research Committee designated a Subcommittee to handle this project. In cooperation with the Illuminating Engineering Society, through the agency of the Joint Steering Committee on Headlight Research, this committee undertook to stimulate and guide experimental research directed toward the determination of the most satisfactory methods of automobile headlighting and, in accordance with the results of research, to formulate a code of recommended practice with respect to head-lamp equipment, adjustment and use. Much valuable information was developed in the course of this joint undertaking, reports of which appeared in the JOURNAL during the years 1925 and 1926.

Manufacturers' Cooperation Enlisted

During these years the major portion of the efforts in the headlighting field were devoted toward enlisting the cooperation of motor-car and lamp manufacturers in a fundamental study of lighting distribution. A suitable instrument for such a study—a four-



DEMONSTRATIONS OF RESEARCH APPARATUS AT THE 1925 ANNUAL MEETING

(Upper Left) Equipment for Showing Fuel Tests. (Upper Right) H. T. Horning Discoursing to an Interested Group. (Lower Left) R. W. Brown and His Early Model Accelerometer and Earliest Model of Recording Device for Measuring Riding-Qualities. (Lower Right) T. S. Sligh, Jr., Demonstrating Bureau of Standards Apparatus for Measuring Dilution of Crankcase Oil

head-lamp test equipment—was developed, its features described, and sets prepared for those who desired to purchase them.

In addition to the foregoing program, headlight research was undertaken at the Bureau of Standards under the guidance of the Society's Headlight Subcommittee with funds provided by the National Automobile Chamber of Commerce. Efforts were made to determine what lighting patterns are best suited to furnish drivers with adequate illumination under the varying conditions of road, atmosphere and opposing lighting. The first part of the program consisted of road tests to ascertain the distance at which an object can be seen under various conditions, supplemented by laboratory experiments. The discrepancies between results obtained on the road and on the test screen led to supplementary tests, making photographic records indicating the area which should be illuminated.

An analysis of these data resulted in certain conclusions regarding beam characteristics which would constitute a satisfactory system of illumination from both the driver's point of view and the point of view of the one who meets it. In October, 1928, a demonstration of lighting patterns, including those developed at the Bureau of Standards, was held at the General Motors Proving Ground at Milford. Mich., before a diversified group of observers.

In the course of further work a photometer for studying and recording glare conditions was developed at the Bureau of Standards. A complete report and analysis of this investigation, it is expected, will be published in the very near future.

Riding-Comfort Investigation

The great importance of the riding-comfort problem was recognized in 1924, and the increasing interest devoted to it by automotive engineers prompted the Society's Research Department to invite a number of members and others known to be interested in the subject to discuss it informally at a supper held in Detroit in January of that year. During the months that followed, the Research Department made a thorough search of the technical literature published in this Country and abroad and compiled a bibliography of more than 500 references dealing with the many phases of the problem.

One of the first instruments for measuring shock due to road contact was the seismograph designed by Dr. Benjamin Liebowitz, while both Dr. Liebowitz and Dr. H. C. Dickinson are identified with early developments in the contact-type accelerometer for measuring the body motion of cars. R. W. Brown and F. C. Mock also were among the pioneers in their work of developing instruments and recording accelerations of relatively high frequency. Later, the Bureau of Standards developed a vibrating chair for testing the susceptibility of persons to vibrations of various periods and amplitudes. Further work was done along similar lines by the late Prof. E. H. Lockwood at Yale University, the Research Department of the Society acting as a medium of exchange of information on the various investigations being conducted.

At the Summer Meeting in 1928, the Research Manager was requested to visit various universities having well-developed physiological and psychological departments, with a view toward interesting an expert in these fields to undertake a fundamental study of riding-comfort. As a result of this survey, the Research Committee was most fortunate in obtaining the services of Dr. F. A. Moss, who is head of the Department of Psychology of George Washington University and a well-known medical doctor in the City of Washington.

Working on the assumption that comfort is the same as lack of fatigue, and using the human body as a measuring-stick, Dr. Moss has devised a whole series of tests for measuring body changes brought about through both muscle and nerve fatigue. Following preliminary riding experiments, he has been able to select a few of these tests which are easily applied and give consistent results in measuring elusive types of fatigue such as are acquired in automobile riding. The development of recording instruments incorporating a counting or integrating arrangement, thus eliminating tedious working-up of records, is an important part of this work, which is under the direction of the Riding-Comfort Research Subcommittee, with funds provided by a group of representative companies in the industry.

In coordination with this program, Purdue University has accepted the task of recording the vibrations of automobile bodies and securing the subjective reactions, orally expressed, of a large group of students. A paper covering this phase of the program, entitled, The Psychology of Riding-Qualities, by G. C. Brandenburg and Ammon Swope, was presented at the Body Session of the 1930 Summer Meeting and is reported in the news account of that session.

When the work has progressed sufficiently, an attempt will be made to correlate the results obtained in the laboratory with the impressions of relative comfort orally expressed by the passengers. Admittedly, human physiology and psychology play a leading rôle in determining satisfaction with a ride. In these experiments, therefore, the human body is being used as a seismograph, with the hope that the reading, though rough in form, may indicate the way in which instruments should be used for measuring riding-comfort. To the automobile manufacturer who is interested in increasing the comfort and luxury of his product, these tests are expected to supply data that will be of considerable value.

Front-Wheel Alignment

The question of the relation between wheel alignment and tire wear was brought before the Research Committee early in 1928, and as a consequence a subcommittee was appointed to get in touch with tire and motor-car manufacturers to obtain from them the information that is available on the fundamentals affecting wheel alignment and to assemble and digest the data.

A survey of the field showed a dearth of experimental data and a considerable difference of opinion on the causes, extent and effects of misalignment. Hence the subcommittee undertook to secure manufacturers' specifications on camber, caster, king-pin inclination, and turning-radius; to check these specifications on the cars as purchased; and to follow up the variation in these factors with mileage, in the hope of ascertaining the effect of wheel alignment on steering, control of the car on the road, and tire wear. Various types of wheel-aligning measuring devices were investigated to determine which would best serve the purpose. Cooperation was offered by the General Motors Proving Ground in securing data on new cars as purchased from the manufacturers and in recording the change in alignment through service on the road.

The data showed that a large proportion of the cars do not check with the manufacturers' specifications at the time of purchase, and indicated that cars do not hold their original settings for any considerable time when driven on the road. However, no correlation between the slight variation in alignment which occurs in ordinary road service, barring accidents, and ex-

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cessive tire wear could be made. The Research Committee is considering the final report of the Wheel Alignment Subcommittee with a view to publishing it in an early issue of the S. A. E. JOURNAL.

Thus, the Research Committee has been called upon to anticipate and meet the needs of the industry as they arose. The relation of gasoline specifications to engine starting, acceleration and fuel economy has been determined. During the last year the vapor-lock problem has been solved, rapid strides toward the goal of a standard method of measuring detonation have been made, and plans have been adopted for conducting an experimental investigation to determine the requisite fundamental characteristics of Diesel fuel-oil.

In the field of highway research, the relative effect on different types of roadway, of various types of motor-vehicle, tire equipment and so forth over a wide range of speed and loading has been investigated. The effect of intensity of headlight illumination, the tilt of the head-lamp, color of the object in the foreground, speed of the car, atmospheric conditions, and other factors which decrease or increase the distance at which it is possible to see when driving at night have been recorded. Instruments for measuring various kinds and amplitudes of vibration have been developed and the mysteries of riding-comfort are being unraveled.

These are but a small part of the sum total of the results of the Society's research endeavors. The service the Research Department has rendered in stimulating pure research, in preventing unwitting duplication of effort and in circularizing the results of contemporary investigations is an intangible contribution, difficult to evaluate but undoubtedly far-reaching in its effect on the perfection of the product and the success of the industry.

Publications of the Society

How All Members Are Kept Informed of Engineering Progress and S.A.E. Activities

THE importance of preserving in some permanent form the papers presented at meetings of the Society and a record of action taken on standards was recognized at the start by the founders of the organization. Therefore, beginning in 1906, TRANSACTIONS OF THE SOCIETY OF AUTOMOBILE ENGINEERS became the depository of the papers and discussions. Published intermittently at first, this soon was changed to a yearly and later a semi-yearly volume that has grown from a 9x6-in. book of about 350 pages per semi-annual volume before the World War to the present 11½x9-in. yearly volume of about 600 pages copiously illustrated. The bound volumes of TRANSACTIONS for the last 25 years constitute an invaluable reference source for investigation of design, research, experimentation and tests in automotive and allied subjects.

Activities of the Society became so numerous by 1911 that the need was felt of reporting them frequently for the information of all the members, hence the S.A.E. BULLETIN was launched as a periodical in April of that year. Three numbers were issued the first year, the April and May numbers containing 16 9x6-in. pages and the October number 48 pages. The present monthly S.A.E. JOURNAL, of 11½x9-in. size containing an average of 140 text pages per month is the outgrowth of that very modest start. To assist in defraying part of the expense of publication, a few pages of paid announcements of makers of materials, component parts and equipment devices were accepted for the S.A.E. BULLETIN beginning in 1914.

Three years later the first number of the JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS succeeded THE BULLETIN. It was of the present size of the S.A.E. JOURNAL and Vol. 1 for the last half of 1917 contained about 400 text pages. In content and form it was very similar to the present S.A.E. JOURNAL, opening with a report of the Semi-Annual Meeting, which was held at the Bureau of

June 4. 1902 THE HORSELESS AGE

For a Technical Automobile Organization.

It has been suggested to us that there is a field for an organization of automobile engineers in the United States, and that the time has come when such an organization could be effected here. The need of such an organization is making itself felt, and there is no reason why it should conflict in any manner with the various other automobile organizations. The National Association of Automobile Manufacturers occupies itself solely with the commercial questions of the automobile business, and represents entirely the commercial end of the industry. On the other hand, now that there is a noticeable tendency for automobile manufacturers to follow certain accepted lines of construction, technical questions constantly arise which require for their solution the co-operation of the technical men connected with the industry. These questions could best be dealt with by a technical society founded, say, on the same lines as the American Society of Mechanical Engineers. The field of activity for this society would be the purely technical side of automobilism, and in other matters it could work in harmony with the clubs and the N. A. A. M. Meetings could be held at specified intervals, at different places, and papers read and discussed on subjects relating to the branch of engineering the society represents.

We shall be glad to accord space to any views on this subject our readers may wish to express.

FIRST PUBLIC SUGGESTION FOR THE ORGANIZATION OF AN AMERICAN SOCIETY OF AUTOMOBILE ENGINEERS, PUBLISHED IN JUNE, 1902

Standards in the City of Washington, and contained papers and discussion, reports of Standards Divisions, including an Aeronautic Division, and several departments, such as Personal Notes of the Members, Book Reviews, Applicants Qualified and Applicants for Membership. The Activities of the S.A.E. Sections were chronicled in 2½ pages in the December, 1917, issue, whereas today each number of THE JOURNAL during the meetings season contains an average of about 15 pages. This reflects the rapid growth in the number and activity of the Sections during the last 13 years.

Publication of Standards and Roster

Standardization has been an important activity of the Society almost from its inception, and the publication of information on standards in a separate reference volume was logical. In 1911 the S.A.E. DATA BOOK was issued. This later became the HANDBOOK, in two loose-leaf volumes, Vol. 1 of standards and Vol. 2 of engineering data. More information regarding the HANDBOOK is given in the article on the History of Automotive Standardization.

The fourth regular publication of the Society is the S.A.E. MEMBERSHIP ROSTER, issued annually since 1908. THE ROSTER for 1910 contained only 14 pages of names and addresses of members, compared with 212 such pages in the ROSTER for 1930.

Constitutional Provisions Governing Publications

Publication of the S.A.E. JOURNAL, the S.A.E. HANDBOOK OF STANDARDS, and TRANSACTIONS OF THE S.A.E. has for years been one of the most important activities of the Society. THE JOURNAL is the official mouthpiece of the Society and carries to members scattered in all civilized countries of the globe the news of all the activities of the organization every month. A large percentage of members in the United States, as well as all those residing overseas, are so located that they cannot attend many National meetings or even Section meetings and have no other means of keeping informed of what the Society is doing than through reading THE JOURNAL.

The Council early recognized, however, that they could keep informed regarding the progress of the automotive industry through trade journals established ten or more years before the Society was established and therefore decided that TRANSACTIONS, and later THE BULLETIN and S.A.E. JOURNAL, should not undertake to record commercial matters but should be restricted to engineering and to the work of the Society. This decision is embodied in the following provisions of the Constitution:

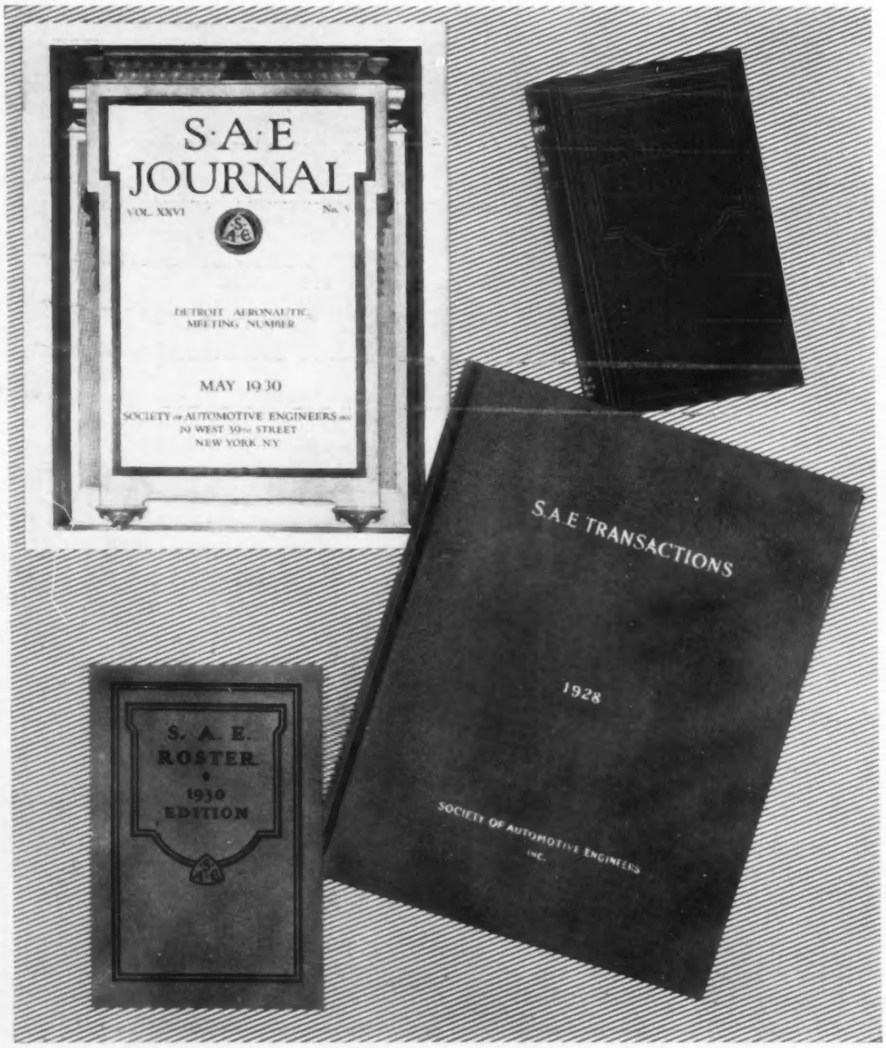
> The official record of papers, reports and discussions and other literature of interest to the Society shall be published and distributed as the Council may direct. . . .
>
> The Society shall claim no exclusive copyright to any papers read at its meetings or any reports or discussion thereon. The policy of the Society shall be to give the papers read before and reports adopted at its meetings wide circulation, with a view to making the work of the Society known, encouraging engineering progress, fulfilling the Society's object and extending the professional reputation of its members. . . . Matters relating to politics or purely to trade shall not be discussed at a meeting of the Society or be included in its publications.

Worldwide Prestige of The Journal

Members of the Council, the Publication Committee and the General Manager of the Society are very jealous of the high professional standing and prestige of the S.A.E. JOURNAL and are constantly watching it to be sure that its quality in every respect is maintained. The publication is confidently believed to have the highest standing of any publication in the automotive field in the United States if not in the world. Frequent requests for the privilege of reprinting papers from it are received from engineering and other periodicals in foreign countries, and a great many papers presented at the meetings of the Society are reprinted in American periodicals.

THE JOURNAL is in a very real sense the members' own publication, bringing to them each month the news of what members of the Society are doing throughout the Country in the National and Section meetings and the activities of the Standardization, Research, and Professional Activities Committees. Of greatest importance

(*Continued on p.* 821)



FOUR PUBLICATIONS NOW PUBLISHED REGULARLY BY THE SOCIETY
The S.A.E. JOURNAL, TRANSACTIONS, S.A.E. HANDBOOK and S.A.E. ROSTER

Powerplant Economics

Piston Displacement versus Horsepower per Dollar

By Alex Taub[1]

Semi-Annual Meeting Paper

Illustrated with Charts

AN ENDEAVOR is made herein by the author to prove by argument and charts based on data that the greatest result per dollar of car cost is obtained by the greatest piston displacement obtainable per dollar expended rather than by the greatest horsepower per dollar.

Maximum result per dollar is a major principle of economics, but horsepower per dollar and piston displacement per dollar are controversial economic fundamentals. The latter is declared to be the accepted principle in the low-price car field, and the author asserts that it should be accepted in the high-price field.

Price class controls the cost of the powerplant, and ingenuity of the engineering and manufacturing departments will control piston displacement. The trends in the different price classes as regards car weight, piston displacement, ratio of weight to piston displacement, and potential and actual performance in the items of economy, durability, acceleration and speed, are shown by charts and discussed.

Adverse effects resulting from the endeavor to increase horsepower per dollar by higher engine-speed are considered and contrasted with effects obtained by a program of seeking maximum piston displacement per dollar.

Rational economics is stated to be incorporated in the latter program because, unlike the program of horsepower per dollar, it does not exclude progress through cooperation of the manufacturing departments with the engineering department to obtain the maximum result per dollar. Possible manufacturing economies to be obtained through such cooperation, in ways outlined, may amount to enough to cover a reasonable increase in piston displacement and, in addition, result in a cleaning up of the engine design.

THE fundamental rule of result per dollar is the major principle of economics on which the future of the automobile industry is founded. It is the only rule that can be applied to an industry whose product reflects a transportation market so complex that every taste, purpose and size of purse is to be met.

Our industry is fortunate in that it is headed by technical merchants; otherwise, our engineering achievement would be slow and progress would suffer.

The industry, as a whole, is dependent on volume; and the extent of the volume is more or less dependent on price in the several classes. A major requirement for successful operation is sustained volume. Sustained volume depends on the reaction of the public to the effort made to meet its demand. This demand is infinite, but must be crystallized in the product offered.

Public acceptance has been sought by price, by platitudes and by merchandising pressure. Today, result per dollar is the one sound factor for maintaining user appreciation.

This ratio of result per dollar varies with price class; the cost goes up as the over-all cost increases. Obviously, the maximum-price class must pay more than the minimum-price class for a unit increase in result.

The powerplant in all price classes must incorporate a true picture of result per dollar, and this rule must apply to user cost as well as to manufacturing cost.

All of these fundamentals have been expounded repeatedly by O. E. Hunt, vice-president of the General Motors Corp., who is perhaps the Country's leading exponent of engineering economics.

To treat the economics of powerplants under the principle of maximum utility per dollar, including every phase to which this can be applied, is entirely beyond the scope of a single paper. Therefore I have decided to apply these principles only to power output.

What is the true picture of result per dollar: horsepower per dollar or piston displacement per dollar?

In the low-price field it is obviously piston displacement per dollar, and I shall endeavor to show that this principle gives the true picture of result per dollar for any price class.

Economic Principles Should Determine Displacement

Analysis of piston displacement by price class brings out the fact that piston displacement does not control price class; neither does price class control piston displacement. Piston displacement should be determined only by economic principles. Price class can limit only the cost of the powerplant; engineering and manufacturing ingenuity must determine the displacement, which should be the maximum per dollar of cost.

Horsepower per dollar is not a matter of simple arithmetic; for example, if a given powerplant costs x dollars and the output is 60 hp., the manufacturer pays $x/60$ per horsepower. However, the public must main-

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tain this powerplant over a maximum number of car-miles, and the cost of this maintenance must be included in the total cost.

Let us assume that the output is increased to 68 hp. without raising the manufacturer's cost. The cost per horsepower is now $x/68$. Does the user obtain the benefit of this? He very frequently finds that he has to make up the difference in horsepower cost by additional upkeep and general operating cost. He probably must consider premium fuels at less miles per gallon because the compression is usually raised into the detonating range. Also, to cover up the decreased low-speed performance, we give him more revolutions per mile, and mixture distribution is usually made uncertain by altering the volumetric-determining factors.

By carrying through a program of increased piston displacement per dollar, we obtain improvement throughout the performance range. It is not necessary to depend on super-efficiency, as we find that reasonable efficiency will suffice. Under this condition we are assured of the greatest number of car-miles of satisfactory performance. Piston displacement is the major factor in the improvement in performance, and, being built into the engine, it will not deteriorate.

One of the most important considerations in a comparison of the horsepower-per-dollar program against the piston-displacement-per-dollar program is carburetion. Large induction systems are a natural part of high-speed engines. Large manifolds are difficult to carburet because of the lower velocity of the mixture at moderate and medium speeds. Precipitation is heavy, and the distribution is usually upset. Regardless of how well the induction system is designed, distribution variation will exist between units of the same product. This variation is a function of distribution sensitivity of the induction system, and the sensitivity increases with the manifold size. Bad distribution, apart from its effect on smooth operation, gives us a choice of one of two major evils: excess heat, due to lean mixture in some cylinders; or excess fuel, to eliminate the leanness.

Troubles Avoided with Larger Engines

Unfortunately, when the horsepower output of a given engine is increased, the waste-heat output is increased. This heat waste is not diminished by additional surface, and the excess must be dissipated through the cylinder-walls and the exhaust valves. Obviously, if lean cylinders exist, the exhaust valves will receive the penalty in the form of still more heat, since leanness creates after-burn. Therefore, in deciding between economy and lean cylinders, durability for the exhaust-valves is generally selected at the expense of economy.

The situation is quite different with the larger engine. For instance, the larger engine can have a limited speed-range as compared with the smaller high-speed engine. The size of the exhaust valves being the same in both engines leaves additional space for cooling the valves in the large engine. The surface-to-volume ratio is raised, since more surface is involved per horsepower; hence the mean temperature of the iron is normal. The additional space for the inlet valves may be utilized by increasing the valve size. Likewise, the timing may be arranged for reasonably high speed.

Under these conditions the engine output would be above requirements; hence we may restrict the speed range by reducing the manifold size below normal for the output desired. The smaller manifold will tend to stabilize induction, which means less variation between like units. It cuts down precipitation; adds ramming effect due to velocity, thereby improving full-load moderate speeds; lends itself to manipulation for distribution correction; and generally improves part-load operation. The smaller manifold also decreases the amount of excess fuel required for acceleration. All these advantages mean better operating economy, as has been borne out by actual operating conditions under which we have been able to definitely determine a marked economy-advantage for cars equipped with relatively large low-speed engines as against cars equipped with smaller high-speed engines.

Sensitivity to performance is a very serious consideration, since the low-price class is predicated on manufacturing volume. An engine that is easy to build good is an absolute necessity; and the high-duty sensitive engine is certainly not easy to build uniformly good. Rapid deterioration of even a small percentage of a large-volume production in the hands of the public would be disastrous. Result per dollar has been the watchword in this price class; and, with this principle in mind, the large engine was selected, thus making piston displacement per dollar the economic factor in the low-priced field.

Factors Affecting Higher-Price-Car Cost

In extending the principle of result per dollar to the higher-price classes, we must consider the major factors affecting price difference and their relation to horsepower per dollar against piston displacement per dollar.

Sustained quality is believed to be the price-making factor. This means that performance shall be sustained for a longer period than in the price class immediately below, that inherent ability to support this sustained performance shall be built in, that smoothness shall be sustained through a longer period and that the silence-creating factor shall be maintained over a longer period.

Sustained performance can be had only by protection of the factors controlling gas-tightness. This means valves, rings and gaskets. I have already pointed out that the low-price-car manufacturer depends upon piston displacement to reduce the punishment of the valves. This same reason applies to piston-rings. In the higher-price class, piston displacement per dollar is just as effective whenever used.

Accuracy is constant for all price classes, since the economics demand it in the low-price field and quality demands it in the higher-price field. Therefore, the higher-price class cannot have higher valve life built in by means of accuracy. Improvement can be made by the use of better heat-resisting valve material. However, valve-seat material in the cylinder-block and degrees fahrenheit are the same for all price classes. Nevertheless, the higher-price class should build in increased valve or valve-seat life.

The exhaust-gas temperature is a very important factor affecting valve life. When this temperature is 1200 deg. fahr., taken close to the valve, no great difficulty is encountered in cooling, provided the ordinary commonsense precautions are taken. However, very little can be done if this temperature is around 1700 deg. fahr. Engines operating with this temperature will deteriorate rapidly. The important item in controlling this temperature is after-burn. This may be caused by lean mixture, poor combustion-chamber design or bad valve-timing.

Disadvantages of High Engine-Speed

I have already discussed the control of lean mixtures and their relation to piston displacement per dollar. I shall now consider timing. High-speed engines require high-speed timing, and this usually means early opening of the exhaust-valve. This timing is a potent factor in raising the mean temperature of the exhaust gas, and, if the valve is closed early so as to keep the valve on the seat for the longest possible time, a high exhaust-residue is trapped in the combustion-chamber. This will reduce the volumetric efficiency and tend to defeat the purpose of the high-speed design. Under some conditions it is possible to find the exhaust-valve inflicted with excess heat from an accumulation of causes, all of which can be mitigated by the larger engine under the auspices of piston displacement per dollar. Obviously, this relief cannot be denied the higher-price cars, since cost stringency does not exist for them to the same extent as it does for the lower-price class. So much for sustained valve-life.

Piston-ring life is affected by almost anything. Excess heat is a factor; therefore the destructiveness of lean mixtures must be remembered. However, rich mixtures have carbon-depositing tendencies, and carbon quickly destroys ring effectiveness. An interesting sidelight on this can be found in Colorado, where ring life is extremely short when insufficient or no compensation is made in the mixture ratio for operation at considerable altitude.

Under the horsepower-per-dollar procedure, with its high mean and maximum pressures and large manifold—hence relatively rich mixture and high waste-heat output—piston-rings operate at an extreme disadvantage. However, the higher-price classes should have sustained ring life, or at least ring life should be extended as compared with that in the low-price field. Undoubtedly, design can play an important part. More rings are essential, and, because of cost leeway, these rings may be fairly thin. The piston design should be such as to dissipate heat as much as possible through the skirt. Aluminum will help; however, aluminum pistons are sensitive to ring action between bore and ring, since this type of piston is fairly open and oil consumption depends on the ring action. Lower speed, lower heat and lower pressure are the most effective elements for ring life. Piston displacement per dollar is the program with this tendency.

Neither piston displacement per dollar nor horsepower per dollar contains many of the elements for good seal through the gaskets. Platitudes cannot be substituted for good judgment. Application of the latter brings success with gaskets. Assuming that the gasket seals are adequate for a given job and that a substantial increase in output is made, the original gasket-seals will be successful in proportion to the amount of reserve seal. Under the increased heat and pressure of a greater power output they may fail if a sufficient margin is not allowed. We undoubtedly will have no difficulty with gaskets if the engine is made larger and the gasket seal is given proper consideration.

The higher-price-car makers can afford to design gaskets of differential thickness to assure the critical spots having adequate material. This undoubtedly would bring about the sustained gasket life to which this price class is entitled. Whether this is their practice is a matter of conjecture.

It is fair to assume from the foregoing that performance-sustaining factors that should exist in proportion to price class can be considerably aided by a program of piston displacement per dollar. Assuming that performance is so sustained, which means that gas-tightness is maintained in proportion to price class, we should find proportionate ability to support this performance. In other words, not only should the bearing size be proportional to the loads, but the factors of safety should be proportional to price class.

Bearing Dynamics at High Speed

The dynamic loads are the most destructive forces that the bearings must sustain. Gas pressure fortunately diminishes with speed; thus the resulting *PV*, or rubbing factor, does not build up very rapidly. The dynamic loads, however, increase as the square of the speed; hence, when combined with rotative speeds, they result in a tremendous build-up of the *PV*'s. This applies to all but one type of engine, the 90-deg. V-eight with crankpins spaced 90 deg. apart. The dynamics to which the main bearings are subjected with this engine are theoretically zero, since the principle of balance necessitates a complete washout of these forces.

We have seen the number of bearings increased during the last few years to offset speed. However, we are all aware that, to be effective, a bearing should have length and alignment. These virtues should represent no hardship to the makers in the higher-price class, where it is assumed that crankcases are normalized so that the bearings maintain their alignment. But bearing length is a space consumer, and, if space is not provided for adequate length, the bearings become steady-rests.

Counterweights are being used to offset the effects of speed on bearing loads, but this means additional cost, although integrally forged counterweights represent a very fair result per dollar. However, there is a practical limit to the proportions of counterweights that can be forged. The result per dollar is low when counterweights have to be bolted on: therefore these are a luxury that rightly belongs in the highest-price field. Counterweights were considered a necessity when top engine-speeds were around 3200 r.p.m. Today we hear about 4000 r.p.m. and higher. Does this mean more counterweights? We hope not. Torsional periods are dangerously lowered by heavy counterweights to a point where, with these speeds, we may meet harmonics heretofore considered beyond the speed range. Torsional-relief mechanisms of today will be inadequate. We no longer have the choice of aluminum or iron pistons under such speeds; they must be of aluminum.

The terrific *PV*'s of the connecting-rod bearings are reaching alarming figures. Dural rods are going to be a necessity if the velocity is to remain. All this is further aggravated by the unfortunate fact that the oil temperature is being forced up by the speed at a time when the maximum lubricating properties are needed.

There can be no doubt that oil-coolers will be as popular as oil-filters in the near future. Meantime, dependence will be placed upon aluminum crankcases and oil-pans, with a maximum ability for air-cooling the oil.

Where Horsepower-by-Speed Quest Leads

What a picture of result per dollar high speed can make. Horsepower per dollar finally leads us to considerably more dollars than horsepower. After the

smoke of engineering achievement has died down, we find that the only thing that is low in proportion is the piston displacement. The general dimensions have increased, and the bearings are large, but the rubbing factors are still larger. The crankcase is aluminum and must be large to cover the structure. The connecting-rods must be strong and yet cannot be heavy, therefore aluminum must be considered. This is a luxurious program that we are thankful to pass on as the prerogative of the high-priced class.

The unspoken resistance to piston displacement on the part of the higher-price class is difficult to understand. It is so obvious that piston displacement per dollar can substantially lower the hazards of horsepower per dollar with the minimum sacrifice of the desirable qualities of any price class. Increased piston displacement is a direct attack on excess dynamics and heat, and our luxury classes would attain the ideal should they combine with piston displacement a moderate portion of the accessories used to offset speed.

Let us look squarely at this demon "speed" by examining its effect on three existing powerplants.

Example *A* at 3000 r.p.m. incorporates a rubbing factor, or *PV*, of 17,377 pressure-feet per minute. This powerplant has iron pistons and steel connecting-rods. This figure sounds rational; however, this powerplant is being operated at speeds in excess of 4000 r.p.m.; and at 4000 r.p.m. the rubbing factor or *PV* is 42,000 pressure feet—a decidedly irrational condition, particularly when we remember that the oil temperature at these speeds is extremely high. The use of dural rods and aluminum pistons will make a reduction in the rubbing factor amounting to approximately 25 per cent, making the factor around 31,000 pressure feet.

Example *B* is another powerplant designed to operate at 4000 r.p.m. The rubbing factor is 64,430 pressure feet. Such a factor is almost unbelievable. It is an obvious case for aluminum rods and pistons. However, even with the modification of 25 per cent, we still have a *PV* factor around 41,500 presure feet. Surely an oil-cooler will be needed here for an additional crutch. At 3000 r.p.m. this *PV* factor is approximately 27,000 pressure feet with iron pistons and steel rods.

Example *C* is still another powerplant that is operated around 4000 r.p.m. The *PV* factor at this speed is 34,000 pressure feet. This can be lowered to approximately 25,000 pressure feet by the application of aluminum. However, at 3000 r.p.m. this *PV* factor would be 14,000 with iron pistons and steel rods, and 11,000 when fitted with aluminum.

Example *A*, with iron pistons, steel rods and its resulting 42,000 *PV*, is a medium-stroke engine. Example *B*, with iron pistons, steel rods and its 64,000 *PV* factor, is a long-stroke engine. Example *C*. with iron pistons, steel rods and its 34,000 *PV* factor, is a short-stroke engine.

Length of Stroke an Important Factor

Despite the fact that the stroke is an important factor, it is not being given sufficient consideration. The crank strokes of the 1930 powerplants are definitely on the long side. How such an important safety factor can be ignored is difficult to understand, as is also the logic of the present tendency toward high engine-speeds, which is leading us to *PV*'s as indicated in the examples cited. What can be the net gain when our engineering advances are absorbed by speed? Instead of engineering advancement, each new development becomes a patch to overcome the speed evil. Aluminum for pistons and connecting-rods is not a preference but a necessity. Instead of additional durability, this material must be used in an effort to stave off disaster.

How much more rational it is to consider the natural advantages of increased piston displacement combined with development of lighter reciprocating parts, thus giving the user the benefit of increased durability, putting into practice the principle of result per dollar. How else can the makers in the higher-price class offer values proportionate in result to their price class? Assuming general adoption of oil-coolers, which would be true economic progress: increasing the speed to the new limitations of the oil-cooler or increasing the number of satisfactory miles of service? An engine speed of 3200 r.p.m. throughout the day will carry a user farther in the same time than 4000 r.p.m., regardless of the miles per hour, particularly if the user fatigue is to be considered.

Elements Affecting Operating Smoothness

We must now deal with smoothness of operation. This virtue most certainly should be found in all price classes, and its constancy should be in proportion to price class. Smoothness may be affected by several elements, beginning with combustion and continuing with dynamics, distortion and engine mountings.

Smoothness is a virtue that needs the yardstick of result per dollar. Its study usually comes after the engineer has had his fling, particularly if his program has been one of horsepower per dollar. Consideration should be given to smoothness at the earliest possible moment in the program.

Absolute smoothness will prevail when there is zero change in load of the engine in its mountings. This would mean zero dynamic residue, zero inertia-torque reaction, zero torque reaction and a dead-smooth combustion. A program of horsepower per dollar will aggravate these elements of roughness with one exception; that is, torque reaction.

Torque reaction usually is associated with larger engines, and there is no doubt that the amplitude of this vibration normally is adversely affected. However, a better understanding of this problem eliminates this argument. We have seen a minor change in the front-end mounting change the resulting torque reaction from a violent vibration between 8 and 14 m.p.h. to a fine vibration from 8 to 25 m.p.h., which is a greater change than can be made by large or small engines. The lightest possible application of rubber will eliminate the long, fine vibration, where no amount of rubber will prevail against the short, violent vibration. Opportunity exists for an interesting study of torque reaction.

All that has been said of dynamics and its effect on bearing life applies with equal force to the problem of smoothness. Increasing the multiple of cylinders used has no doubt created new standards of smoothness; but more has been done than to increase the number of cylinders. The top speed of the engines has been raised to a point where at high speeds the dynamic roughness is about what it was before the increase in the number of cylinders. I particularly refer to the popularizing of the eight-cylinder engine. However, increasing the number of cylinders is a most logical method of establishing the price-class difference for smoothness. By this route it has been possible to decrease the bulk of the charge per cylinder, thus automatically reducing

the heat output per individual cylinder. We extend thereby the life of pressure-maintaining elements. This, of course, assumes the original r.p.m., which in most cases today does not apply.

Program Aids Eight-Cylinder Smoothness

Greater effort is being made today to obtain the maximum horsepower per cubic inch for the eight-cylinder engine than was ever made with the six. The ability of the eight to establish and maintain its additional smoothness is soon lost unless we apply to it the principle of piston displacement per dollar. The fact that the eight-cylinder charge is two-thirds of the six-cylinder charge offers sufficient leeway per cylinder to warrant consideration of an increase in displacement, so that we may be permitted to use the ability of the eight to run smoothly. Additional cylinders is a better answer than additional power from the same unit.

Combustion control is an important economic factor, for herein lies a means of partly mitigating the evils of high efficiency. That combustion roughness must be reckoned with in any smoothness program is a well-established fact. This type of roughness, when coupled with crankcase weakness, is known as "thump." When roughness without thump is evident, it has an unpleasant harshness. The major cause of this evil is the rate of combustion, or the acceleration of the pressure rise. The more rapid the rate change, the rougher is the result. Combustion control can bring about a constant rate of pressure change; however, if the rate of change is rapid even though constant, harshness will ensue. The most important factor affecting the over-all rate is the compression ratio, on which engineers have depended to obtain efficiency. Recently there has been a marked tendency to lower this ratio because of the ensuing roughness. It is a discouraging project for even the makers in the maximum-price class to build into the crankcase sufficient rigidity to withstand the effects of the combined evils of high-speed dynamics and combustion roughness.

Extreme compression-ratios are not needed with increased piston displacement, and since the speed may be lower the result is more in keeping with engineering ideals, particularly if the same effort to secure combustion control is applied to the larger engine with the lower compression. Spreading the additional volume through the combustion-chamber without affecting the rate of burn is not difficult. However, it is impossible to add to the compression ratio without raising the rate of burn. To be sure, combustion-chambers are in use today that are smoother at a compression ratio of 5.5:1 than others would be with a ratio of 4.5:1. In spite of this fact, it is better for the builders in the higher-price classes, who must effect absolute smoothness, to consider the best combustion-chamber principles applied to the increased volume.

Smoothness by price class can be and is established by combining, according to price-class needs, the relatively large engine with the number of cylinders used. Add to this the best in thermodynamics, and we have a maximum result per dollar based on piston displacement per dollar.

Noise Increase with Engine Speed

Noise is and should be inversely proportional to price class. However, noise increases directly with speed. Therefore the means to obviate noise is proportionately difficult. Noise is a major hurdle that must be met in any program of horsepower per dollar. The elimination of noise may constitute the item of cost that serves to make the horsepower-per-dollar ratio unsatisfactory; in fact, it usually raises the cost to a point where the additional horsepower obtained by more speed is more than offset by the cost increase.

Engine noises that must be dealt with are resonance and those caused by valve mechanism, exhaust and air intake. These are products of speed, and their elimination jeopardizes the result-per-dollar ratio.

Admitting that the builders in the highest-price class have done splendidly with these problems, nevertheless the same effort applied to a larger but slower engine would have given a much better result per dollar.

Crankcase resonance is critical and is usually set up by a transverse bending of the crankcase at its center. This movement is caused by the dynamic forces, which are proportional to the square of the speed. A moderate change in speed may be the difference between a satisfactory and an unsatisfactory condition. Stiffening of the crankcase is the answer. However, the business of material distribution in a crankcase is a major study in economics. We have found surprisingly little increase in crankcase rigidity between crankcases that have considerable difference in weight. Yet we have seen 3 per cent in weight added to an original design produce 40 per cent additional rigidity.

The larger engine operating at slower speeds, partly treated for stiffness, will give the maximum result per dollar.

Nothing need be said with regard to the valve mechanism, since the evils of speed are obvious here and the difficulties of overcoming them are well known. It is also obvious that the program of piston displacement per dollar will help.

Exhaust and intake noises are tied up to a certain extent with high-speed timing. Noise due to increase in cylinder volume can be reduced by decreasing the velocity in the exhaust manifold and pipes. But noises caused by timing must be remedied by timing, and this usually means compromising the high speed.

For minimum noise with increased power, piston displacement per dollar is the program that leads to result per dollar.

Trend by Price Class Analyzed

Analysis of the trend among price classes is an interesting study. Fig. 1 is a comparative chart of weights and piston displacements. Five price classes are represented. Each price class is represented by the average of six or seven different makes of car. Thirty different makes of automobile are included in these averages, and great care has been taken in the preparation of these data. The information does not represent the thoughts of any single organization, but is a summation of the thoughts of an entire industry by price class. From Class 1 to Class 5 we find the consolidated thoughts of the industry for 1930.

The curve *A* in Fig. 1 represents weight by price class. Note that the weight increases in a straight line for price classes from 1 to 4, and that the highest-price class has the maximum weight increase. Curve *B* represents piston displacement by price class. Note that the displacement increases in proportion to weight. Curve *C* represents the pounds of car per cubic inch of piston displacement. It is interesting to note that the average for this combination is similar for the highest and lowest price-classes, that is, 14.4 lb. per cu. in. of dis-

placement. The second price class has the highest weight-to-displacement factor; therefore, from our viewpoint, the lowest rating in result per dollar. The third price class is slightly better, and the fourth price class is good. An ideal situation for the industry would be to have the displacement curve *B* follow the weight curve.

Potential Durability, Economy and Performance

Fig. 2 presents a comparison of potentials by price class. These data are also the averages for each price class and therefore representative.

Curve *A* in Fig. 2 represents the potential performance in cubic feet of piston displacement per ton-mile by price class and, being made up of physical facts, gives a true picture. The characteristic of this curve indicates that the trend is upward from the lowest to the highest price classes. However, the second price class is relatively low. This price class indicated poor weight-to-displacement ratio. The third price class shows high potential performance despite a poor showing in weight to piston displacement, which is a very suspicious combination. Price classes 4 and 5 are apparently in line, as was their weight to piston displacement.

Curve *B* represents the potential economy in cubic feet per car-mile by price class. These data likewise are made up of physical facts. The trend is downward from the low to the high price class, as was to be expected; however, the drop could be improved. Note that Class 3, which is under suspicion, is out of line on economy potential.

Curves *C* and *D* represent the potential durability, and the factors here are made up of physical facts. They are revolutions per mile per hour and piston travel in feet per car-mile. Obviously, the lower the factor the higher the durability potential. The lowest-price class incorporates the highest durability poten-

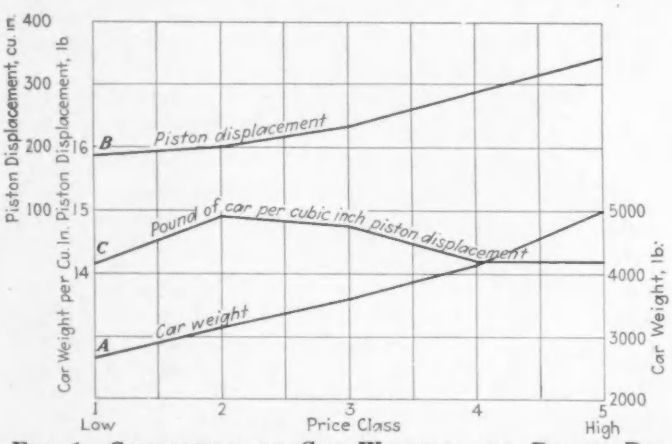


FIG. 1—COMPARISON OF CAR WEIGHTS AND PISTON DISPLACEMENT IN FIVE PRICE-CLASSES

tial. This is to be expected since the builders in this price class are economy-conscious and are making a real effort to obtain result per dollar.

Price Class 2 indicates the lowest durability potential and is also low in performance potential. The reason for this is indicated in Fig. 1, in which this class showed the maximum number of pounds per cubic inch of displacement. This price class does not compare well for result per dollar. An increase in piston displacement would be advantageous.

Price Class 3 is only slightly better than Class 2. It is obvious that this price class has made a questionable sacrifice of durability potential for performance potential.

The durability factors of the Classes 4 and 5 are

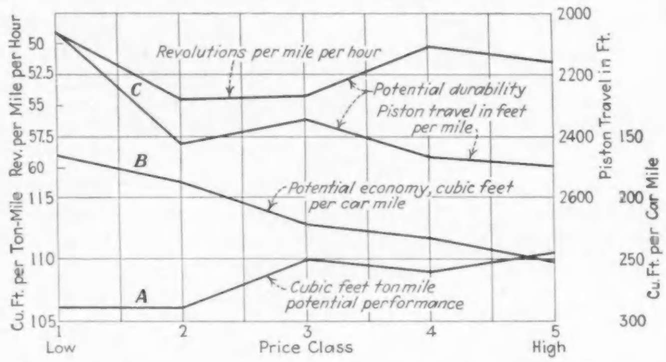


FIG. 2—COMPARISON OF POTENTIAL DURABILITY, ECONOMY AND CAR PERFORMANCE BY PRICE CLASSES

more or less of a compromise. The revolutions-per-mile factor shows up well. However, there is evidence of another evil—long stroke—which has lowered the durability by increasing the piston travel. This is something worth thinking about.

Actual and Potential Acceleration Compared

Fig. 3 is extremely interesting. An effort is made here to call a spade a spade. This chart gives the comparison of the potential performance and the actual measured performance of these cars. The dash-line curve represents the potential performance by price class, and Curve *A* represents the average of the actual accelerations from 10 to 25 m.p.h. Price Classes 1, 2 and 3 are very much alike. Class 3 in slightly lower than Classes 1 and 2, in spite of the fact that potentially durability was sacrificed for potential performance in this price class. It must be remembered that we are not discussing a single make of car but an entire price class. Does this mean that these classes are running "soft" at the low end so as to improve durability? Would it not be better to increase the piston displacement and thereby balance things?

There is a considerable increase in acceleration from 10 to 25 m.p.h. in Classes 4 and 5, with Class 4 leading.

Curve *B* is the actual average acceleration from 10 to 35 m.p.h. We find here a marked improvement in Classes 2 and 3, indicating that the poor showing at 10 to 25 m.p.h. is brought about by a lack of low-end performance, due no doubt to the lack of piston displacement.

Curve *C* is a hill-climbing factor made up from the reciprocal of the actual time required to climb a given hill with a given start. Our first difficulty in averaging was experienced in making up this curve. The lowest price class incorporates a maverick. One of the makes included in the average of this class is a very bad hill-climber. Although its potential is the highest in its class, its poor actual hill-climbing ability ruins the comparative of the class average. Without this one make, the average is about equal to that of Class 2.

Classes 2 and 3 are again relatively low, with Class 3 making an extremely bad showing as compared with its potential, indicating again that the sacrifice of durability potential for performance potential has been in

vain. Classes 4 and 5 are well up in actual hill-climbing ability. The highest-price class, however, does not compare favorably with Class 4, indicating a comparative lack of low-speed performance between these higher-price classes.

Real and Possible Road Economy Compared

Fig. 4 represents a comparison between actual and potential road economy as indicated by the broken line. Curves *A*, *B* and *C* represent actual average road economy by price class at 20, 40 and 50 m.p.h.

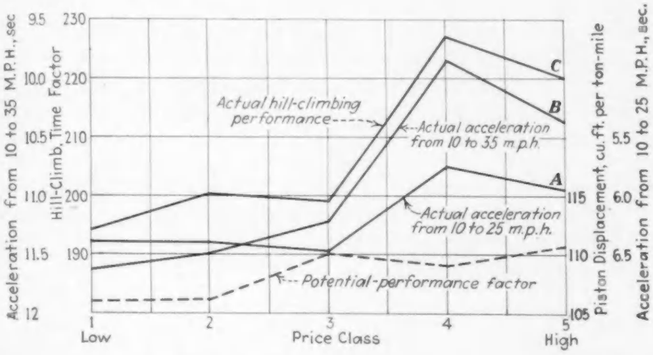


FIG. 3—COMPARISON OF ACTUAL AND POTENTIAL ACCELERATION AND HILL-CLIMBING PERFORMANCE BY PRICE CLASSES

The lighter cars of the lower-price class show the best road economy in spite of the fact that the same maverick encountered in the hill-climb tabulation exists for economy. Note the difference in trend between the potential and actual economy for Class 2. This price class has the highest pounds of car per piston displacement, or relatively the smallest engine, yet the economy is poorest in spite of any allowance we may make for weight. This general condition has been observed whenever undersized engines are used.

The actual economy drop of Class 3 is fairly proportional, as is also that of Class 4, although at 20 m.p.h. this class falls off more rapidly than it might.

The highest-price class is extremely poor on economy. The prerogative of the builders in this class has always been to ignore road economy and, from the results as indicated, they are still exercising it, but the time is not far distant when they will give heed, not because of upkeep cost but for cruising radius unless these vehicles are to join the tank-car class.

Program Means Larger Engine at Same Cost

A program of piston displacement per dollar is the soundest program an engineering department can face. Not only does it bring us engineering ideals, but it has a wholesome effect on the department as a whole. Piston displacement per dollar does not necessarily mean a larger engine with thinner walls, but a larger engine at the original cost of a smaller unit. This program sends the engineer into the highways and byways of the production organization. He goes to the foundry in an attempt to determine what makes up the cost of iron per pound. The engineer knows that herein may lie a major portion of his gain in piston displacement per dollar. If he can reduce his price per pound, he can have the very few pounds necessary for the increase in size.

Castings in the foundry are inclined to collect cost barnacles resulting from various changes during their production life. These barnacles are forgotten but they constitute an item of cost. Sometimes cores that are difficult to make or set have crept in during the pressure of production and require an extra inspection. Perhaps a wall is being cast much thicker than is necessary because existing construction does not permit of proper ribbing. Piston displacement per dollar requires that such superfluities be ferreted out and eliminated.

The difference between the cost of iron at the cupola and the cost to us is made up of a definite cost for material and labor. The cost per pound is reduced by eliminating operations, cores, inspections and foundry hazards.

Engineers have been prone to overlook the possibilities of making foundry studies while the new product is still liquid. Foundrymen have been satisfied to demonstrate their ingenuity in overcoming a difficult problem in molding. They should, instead, return the design to the engineer to be cleaned up so as to permit the use of correct foundry principles, utilizing the maximum of familiar practice. This means a minimum hazard and hence lower cost.

Consideration should be given to the elimination of expensive driers in the foundry, and this can be accomplished usually by slight changes in design. Complicated driers are expensive to make and are perishable, as they will warp and must be replaced. They form part of the cost.

Interesting results would be obtained by engineers if they requested their foundries to send them a list of driers and suggested changes for their elimination. They would discover how much trouble an insignificant fillet or projection may create.

Economies in Forging and Machine-Shops

A visit to the forge shop is also in order, to determine what our forging costs cover. Can the scrap cost be cut? Can the dies be simplified and the die life be enhanced by redesigning the forgings? Open-minded investigation of forging conditions will not fail to lower the cost. Cost barnacles exist in the forge shop as in the foundry; and the forge man, like his brother in the foundry, is prone to expend his ingenuity on overcoming a difficult job, thereby creating a hazard which

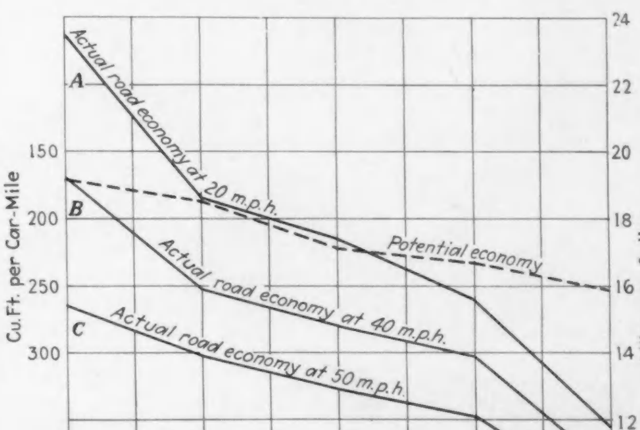


FIG. 4—COMPARISON OF ACTUAL AND POTENTIAL ROAD ECONOMY AT 20, 40 AND 50 M.P.H. BY PRICE CLASSES

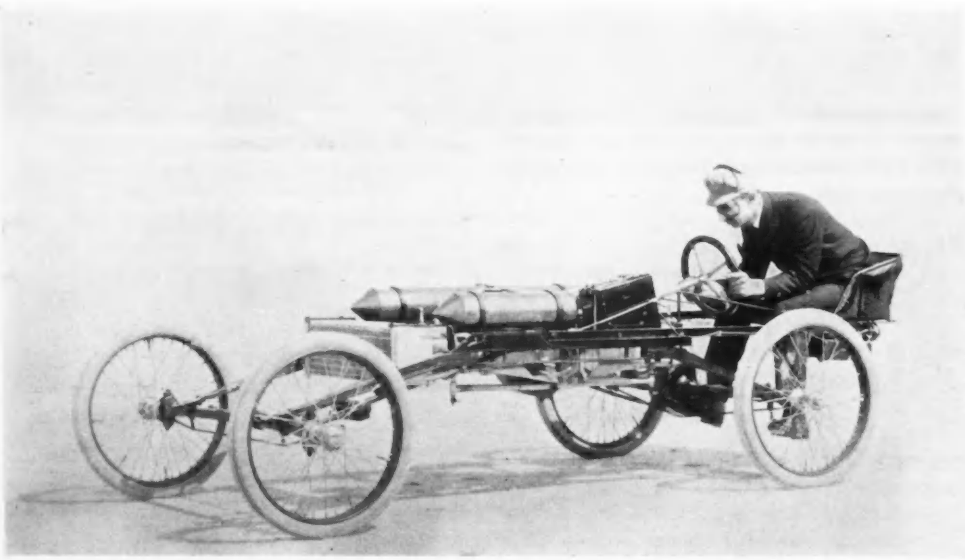
usually finds its way into the cost. Ingenuity is most profitably expressed on paper, and it behooves the producing part of the organization to insist that early consideration be given its particular problems.

The machine-shop is also a fruitful source of cost reduction. We may find here a multitude of individual operations that, with a little consideration, can be combined with others. An interesting list would be obtained if the engineer repeated the listing of uncombined operations on his current product. Real progress against cost can be made with such information.

The most economical drill size to be used should be given consideration by the engineer. His machine-shop man will tell him a drill of a certain size gives the best combination of speeds, feed and first cost. For instance, consider the fact that drilled connecting-rods are becoming popular. Examination of them brings out the fact that a 7/32-in. hole is fairly universal. Does this mean that the 7/32-in. drill is the most economical to use? It does not. One organization started with this size and others have followed. This is a fairly costly hole to drill, yet with proper consideration of drill size this operation is not at all prohibitive. Mention is made of the drilled connecting-rod because it serves to illustrate a principle that should be applied to every drilled hole in the powerplant.

A summation of the possible economies to be obtained by these investigations may equal a cost large enough to cover a reasonable piston-displacement increase, and, in addition, we have had a housecleaning in matters of design.

The purpose of this paper is to place before you a case for piston displacement per dollar. I have endeavored to prove by argument and chart that result per dollar is exemplified in piston displacement per dollar. I have tried to show that rational economics are incorporated in this program; rational because a program of horsepower per dollar is a purely engineering program that excludes progress by departmental cooperation, except that the hazards are passed on to the manufacturing organization. Piston displacement per dollar demands cooperation of all departments in an organization to obtain the maximum result per dollar and, therefore, represents a substantial and healthy program.



R. E. Olds 1903 Pirate in Which H. T. Thomas, Shown at the Wheel, Broke Alexander Winton's Straightaway Record of 68 M.P.H. on the Daytona Beach at the Terrific Speed of 86 M.P.H.

Comment on American Passenger-Car Gearsets

By Herbert Chase[1]

Semi-Annual Meeting Paper — *Illustrated with* Drawings

Modern transmissions are discussed and criticized primarily with respect to ease of gearshifting, quietness and relative simplicity. The advantages and disadvantages of four-speed gearsets as compared with the three-speed type are set forth and the several types of the former are illustrated and described briefly. American automobiles on the market are divided into five classes according to the relation of engine size and speed, type of transmission and gear reduction in the axle. This variety is said to show that a marked difference of opinion exists among engineers as to the best combination of these variables.

Seven advantages claimed for four-speed gearsets, in addition to ease of gearshifting and quiet operation are listed, but question is raised whether all can be realized in practice and at what penalty and whether they cannot be obtained with some simpler form of construction. The subject is analyzed in the paper on this basis.

The author draws no final conclusions but leaves the impression that most if not all the alleged advantages of four-speed transmissions can be obtained with the simpler, less expensive three-speed type that requires less gearshifting. He tells what has been achieved by three-speed advocates in respect of quiet operation, easy shifting and longer car life without sacrifice in general performance.

Suggestions are made that more needs to be done toward reducing the ratio of car weight to engine power, that helical gears present possibilities that should be investigated and that various methods of insulating the gearset to reduce the transmission of vibrations will help to reduce noise.

Automotive engineers apparently are agreed that in at least two respects gearset design is in need of improvement; namely, those which affect ease of shifting and those controlling noise in operation.

Most major changes in the design of gearsets within recent years have been directed, at least in part, toward these ends, but other changes of a rather more radical nature and about which a great diversity of opinion exists have entered. They have complicated the whole situation because they have distinct bearing upon the type and size of engine to be employed, the rear-axle gear-ratio, the design of other units such as the clutch, propeller-shaft and universal-joints, the weight of the car, and, last but not least, upon salability of the product. I refer to the number of speed changes available.

A trend toward four-speed gearsets has developed, but whether this change has been dictated more by expediency to meet a particular sales situation than by purely engineering considerations is a moot question.

Undoubtedly part of the automobile-buying public believes, or can be led to believe, that four speeds necessarily are better than three, just as many persons have come to rate cars roughly in proportion to the number of cylinders the engine has. It may be good business to take advantage of this situation, but my intention here is to discuss the purely engineering aspects of the subject.

At the moment there are at least five general classes of car on the market:

(1) Cars combining a relatively large but comparatively low-speed engine with a three-speed gearset and a small axle-reduction

(2) Cars incorporating a moderately small, relatively high-speed engine, a small axle-reduction and a four-speed gearset intended to keep the engine running at low speed most of the time

(3) Cars incorporating a moderately small, relatively high-speed engine, a three-speed gearest and a rather large axle-reduction

(4) Cars having a medium-size engine of fairly high speed, a four-speed gearset and a rather large axle-reduction

(5) Cars combining a large, relatively high-speed engine, a three-speed gearset and a rather large axle-reduction.

Other classifications doubtless could be added, but this variety is great enough to indicate that a marked difference of opinion exists among engineers as to the best combination of the variables mentioned. The power-weight ratio of the complete car is a factor of importance in determining the combination selected but it is not always the controlling factor.

Advantages Claimed for Four-Speed Gearsets

Since the four-speed gearset has made a place for itself in 15 or more makes of chassis and, in certain cases, on several models of the same make, its construction is deserving of comment and the advantages claimed for it demand analysis. Its disadvantages also should be weighed.

All four-speed types in regular use on American cars drive direct only on fourth speed; none are over-geared. In all cases efforts have been made to assure a quiet third speed. Four makers (the Warner Gear Co., the Detroit Gear & Machine Co., the Chrysler Corp. and the Warner Corp.) aim to achieve this relatively quiet operation by the use of internal-gear trains. The Pierce-Arrow Motor Car Co. employs herringbone gears

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in constant mesh, and the Packard Motor Car Co. uses spur gears that are carefully ground and fitted. Of these makes all except the Packard have certain design characteristics intended to make possible, and presumably achieving, an easier shift from third to high speed and vice versa than is obtained in shifting from second to high in conventional three-speed gearsets.

These advantages of easier shifting and quieter operation in next-to-top speed appear to have been attained, but some engineers of standing still have to be convinced that operation is any less noisy than that of well-made conventional types. Quiet operation and easy shifting can be and are realized in some three-speed gearsets to the same extent as in the four-speed variety; they are not advantages peculiar to the four-speed variety, although more frequently made available in that type at present.

Other advantages said to be realized by American four-speed gearsets now in use include:

(1) Lower engine-speeds, with consequent decrease in engine noise and vibration, especially at high car-speeds, and less fatigue in driving
(2) Fewer engine revolutions per mile, with resulting decrease in engine wear and friction losses
(3) A smaller reduction in rear-axle gears, which consequently are smaller, lighter, cheaper and quieter
(4) Lower propeller-shaft speeds, resulting in less whip and noise, or permitting the use of lighter and less expensive parts
(5) Improved fuel economy, resulting from a better power-factor, and lower oil consumption
(6) Ability to accelerate more rapidly without excessive gear noise in the next-to-top speed
(7) Shorter steps between gears and a lower low gear for emergency use
(8) General excellent performance in third speed in traffic.

This is a rather imposing list of advantages, but can they all be realized in practice, and, if so, what penalty is paid? Or can they be obtained with some simpler form of construction? An analysis should bring us to the crux of the matter.

Sacrifice Must Be Made for Acceleration

With an engine of given size and small reduction in rear-axle gears, a lower average engine-speed can be realized with a four-speed gearset, engine noise and vibration can be decreased thereby, and it is fair to assume that less fatigue will result on this score. There also will be a consequent saving in engine wear, reduced friction losses and higher fuel and oil economy.

These are very real advantages, but to obtain them a definite sacrifice is what is loosely termed "performance" in top speed is necessary. The car no longer has the same ability for rapid acceleration unless a shift is made to the quiet third speed. If the driver is willing to make the shift, well and good, but it must be made as often as the car is accelerated if the rapid rate of acceleration to which American motorists are accustomed is to be realized.

Is the average driver willing to make this sacrifice? Unless he is, his car will prove somewhat logy as compared with corresponding cars having greater axle-reductions. As I see it, the fate of the present type of four-speed gearsets hinges largely upon the answer to this question. Apparently the answer has been given to the satisfaction of at least one car manufacturer, for, starting with small-reduction axles when the four-speed gearset first was applied, this company soon went back to the larger reduction but retains four speeds. Other companies continue to apply axle gears giving a large reduction.

Engineers with another viewpoint say in effect, "Get both the rapid acceleration and the advantages of a low-speed engine and .mall axle-reduction, without going to four-speed gearsets, by the simple expedient of using a larger engine." This has some obvious advantages in the way of simplicity, but in the large-car group it is presumed to involve the use of a larger and heavier engine and driving parts and, consequently, a heavier and more expensive car; also, possibly, an increase in the number of cylinders if reciprocating parts are to remain as light as is necessary for the degree of smoothness desired.

Axle Ratios with Four-Speed Gearsets

It is a significant fact, however, that both the Ford and Chevrolet companies follow this practice and succeed in producing cars of light weight and low cost. Another example of similar practice but applied to a larger chassis is the Studebaker Commander Six, which uses a 3.7:1 rear-axle ratio, yet is reported to perform well on hills and during acceleration. This car has a

TABLE 1—REAR-AXLE RATIOS OF AMERICAN CARS WITH FOUR-SPEED GEARSETS

Car Make and Model	Piston Displacement of Engine, Cu. In.	Maximum B-Hp. at Specified Speed	Make of Gearset	Standard Rear-Axle Gear-Ratio
Blackhawk L8	241.5	85 at 3200	Detroit	4.5
Blackhawk L8	268.5	88 at 3100	Detroit	4.5
Chrysler 70	268.4	87 at 3200	Own	4.1
Chrysler 77	268.4	87 at 3200	Own	3.82
Dodge Senior Six	241.4	61 at 3400	Warner Gear	3.77
Durant 6/17	248	70 at 3000	Warner Corp	3.72
Elcar 95-96	246.7	90 at 3000	Warner Gear	4.9
Elcar 130-140	322.2	140 at 3300	Warner Gear	—
Franklin 145	274	95 at 3100	Detroit	4.54
Franklin 147	274	95 at 3100	Detroit	4.25
Gardner 136	185	70 at 3500	Warner Gear	4.45
Graham Special Six	224	76 at 3400	Warner Gear	3.91
Graham Special Eight	298	96 at 3400	Warner Gear	3.9
Graham Custom Eight (127-in. wheelbase)	322	120 at 3200	Warner Gear	3.64
Graham Custom Eight (137-in. wheelbase)	322	120 at 3200	Warner Gear	3.9
Kissel 8-95	246	95 at 3400	Warner Gear	5.1
Kissel 8-126	298	126 at 3600	Warner Gear	4.8
Jordan Z Speedway	322.2	114 at 3200	Warner Gear	—
Marmon Big Eight	315.2	125 at 3400	Warner Gear	4.81
Packard 726	320	90 at 3200	Own	4.37
Packard 733	320	90 at 3200	Own	4.67
Packard 740	384.8	106 at 3200	Own	4.37
Packard 745	384.8	106 at 3200	Own	4.37
Peerless Master Eight	322	120 at 3200	Warner Gear	4.45
Peerless Custom Eight	322	120 at 3200	Warner Gear	4.45
Pierce-Arrow 132	340	115 at 3000	Own	4.42
Pierce-Arrow 125-139	366	125 at 3000	Own	4.08
Pierce-Arrow 126	385	132 at 3000	Own	4.42
Stutz Series M	322	113 at 3300	Detroit	4.5
Windsor 8-92	268.8	86 at 3200	Warner Gear	3.9

348.3-cu. in. engine said to develop a maximum of 75 b-hp. at 3000 r.p.m. The only other American cars I find listed with standard axle-ratios below 3.85:1 are the Dodge Senior Six (3.77), Chrysler 77 (3.82), Chrysler Imperial (3.77), Durant 6-17 (3.72) and Graham Custom Eight 3.64). All of these (except the Chrysler Imperial) have four-speed gearsets.

The standard gear-ratios listed for all American cars using four-speed gearsets are given in Table 1, from which it appears that the average reduction is not greatly different from that on most cars with three-speed gearsets. One natural inference to be drawn from this table is that, despite apparent possession of advantages (1) to (5) given in the preceding list, the disadvantage of more frequent gearshifting required with four-speed gearshifts and small axle-reductions has led those who use the four-speed gearset to sacrifice these advantages and employ almost if not quite as large reductions as commonly are used with three-speed gearsets.

Two of the remaining advantages claimed, (6) and (8), are not admitted by some engineers. They do not deny that the four-speed gearset does permit of more rapid acceleration in third speed than the usual three-speed arrangement gives in top speed, but some say that, as regards the noise under these circumstances and the general performance in traffic, there is little if any improvement over conventional three-speed arrangements. Others profess to find considerable improvement in both directions. As usual, where precise measurements are difficult or impossible, a difference of opinion is more than likely.

Advantage (7), shorter steps between gears and lower low gear for emergency use, is conceded but carries much too little weight to be of substantial moment if the other advantages claimed are not conceded. Ordinarily, the conventional low gear is quite low enough for all practical purposes and the steps are not so far apart as to warrant the added complication and more frequent shifting likely to be associated with four-speed gearsets.

Complexity a Drawback

On the negative side of the argument certainly must be included the added complexity of the four-speed gearset itself. No more than a glance at the accompanying designs of three-speed and four-speed transmissions is needed to determine which type is the simpler and the easier to manufacture. The four-speed sets, and in particular those using internal gears, present some difficult and costly operations in manufacturing and also in service when repairs or replacements are required. Some problems of the same or a similar character must be faced and solved if quiet operation on other than direct speed is to be obtained in three-speed sets, but, if such a quiet set is compared with a four-speed internal-gear type, the latter is found to be noticeably more complicated.

A criticism of the four-speed gearset that seems to me to be fair although somewhat severe is contained in the following summary by Delmar G. Roos, chief engineer of the Studebaker Corp., contained in a paper he presented before the Indiana Section of the Society on Feb. 14, 1929:

> Anyone manufacturing the ordinary three-speed type of gearbox and trying to make a good job of it knows that he has his hands full. Anyone who examines the best designs of the four-speed gearbox with the internal third gear knows that to put a transmission of this kind in the plant and manufacture it with any degree of uniformity is certainly going to be one of the most difficult problems that has yet been put into an automobile factory.
>
> It is not quite clear to the disinterested observer just what people expect to get from the four-speed gearbox. If the engine size is adequate to permit the use of a rear axle of 3.7 to 3.3:1 ratio, such as Studebaker used in its Commander, with excellent performance on hills, straightaways and acceleration, it seems that very little could be hoped for from putting a four-speed gearbox in such a car. On the other hand, if the engine were inadequate so that the rear-axle ratio, to give satisfactory performance on hills, would have to be say 4.5 or 4.8:1, we can see that, by putting a four-speed gearbox in, we might get an axle that was 4:1 or even 3.9:1 on high and having a ratio in third speed of perhaps 5.5:1.
>
> The question boils down to this: Can the problem be better and more simply met by an engine of larger displacement, economically designed as to weight, rather than an engine of intermediate or small displacement in proportion to the weight of the car?
>
> Our experience so far in driving cars equipped with these gearboxes, particularly in traffic in metropolitan districts and in mountainous districts, has left us at a loss to find that it aroused any particular enthusiasm. They are invariably noisier in third speed than any gearbox that our customers have been willing to accept. Our opinion is that the whole issue is a false one which has been raised in a desperate effort to secure attention and that, unfortunately, it may take hold simply as a fad; but we do not believe it has any sound foundation to rest on and, in the long run, the public is going to pay, just as it always does, for the experimenting of the engineers.

Reduction of Engine Speed

One justification often advanced for the four-speed gearset is that it keeps the engine at a speed that is lower, more comfortable and much less destructive when the car is traveling at high speed than does the three-speed set. This is true when a small reduction in axle gears is used, but not otherwise. Moreover, it should be emphasized that relatively few car users do any large part of their driving at speeds above 50 m.p.h., even though there is perhaps a tendency in that direction in localities where excellent roads and long level straightways abound. Those who do drive much at high speeds may perhaps choose the car with four-speeds and small gear-reduction and put up with extra gearshifting under some conditions. I venture to say, however, that fully 95 per cent of American drivers would not choose axle ratios much below 4:1 if they understood the alternative and what it involves.

In the foregoing the references have been confined to four-speed gearsets driving direct in high. A good case might be made for over-gearing in fourth speed, although that, too, presents some disadvantages. I shall not attempt to go into that question here, but it may be said, in passing, that a really quiet over-geared fourth speed intended only for use in making high road-speeds, combined with axle-gear reductions that are not too small, would tend to keep the engine at more acceptable speeds when the car is driven at high speed. This need not involve such frequent gear-changing as the direct fourth with axle reductions of about 3.8:1 and engines of moderate power.

Differences in Four-Speed Gearset Types

So much for present-day four-speed gearsets as a type. Differences between types may be discussed

along with those of three-speed types, for, as far as details of design are concerned, many of the same factors are involved, assuming that the aim in general is to produce quiet operation together with easy shifting.

On the score of simplicity, the Packard spur-gear set, shown in Fig. 1, seems to lead the four-speed types, for it has no more bearings and only one more spur gear than conventional three-speed types, which it resembles closely in other respects.

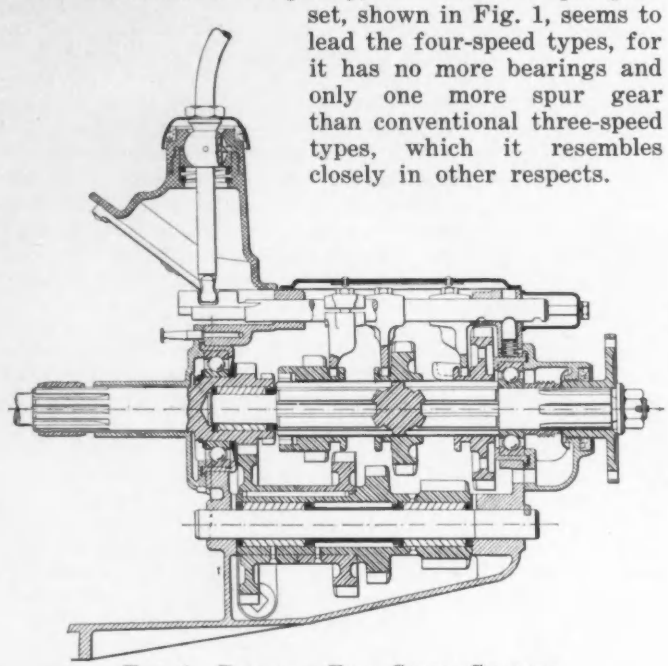


FIG. 1—PACKARD FOUR-SPEED GEARSET

Only Straight Spur Gears Are Used in This Set

Next in order is the Pierce-Arrow gearset, illustrated in Fig. 2, which uses herringbone gears for the constant-mesh and third-speed pairs. The number and arrangement of gears and bearings in this gearset follow closely the design of three-speed sets, except that the extra speed requires one additional gear on the main shaft, a pair of roller-bearings for mounting this gear and a toothed clutch for engaging it with the main shaft. The herringbone gears are cut in two separate blanks afterward riveted together. Those in the constant-mesh pair are all 12-pitch, but those in the third-speed pair are 12-pitch for the forward pair of blanks and 10-pitch for the rearward pair, an arrangement intended to promote quiet operation by breaking up any synchronous hum. The slight additional complexity as compared with a straight four-speed spur gearset is offset by the quieter operation in third speed.

[2] See THE JOURNAL, February, 1927, p. 250.

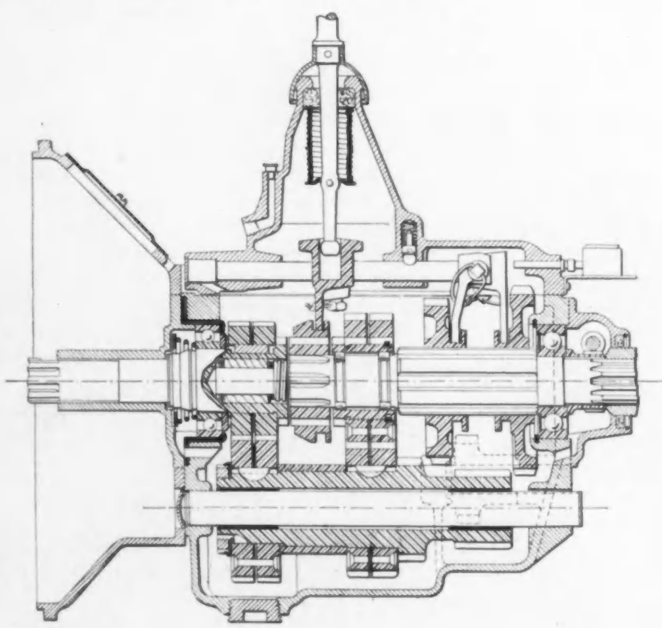


FIG. 2—PIERCE-ARROW FOUR-SPEED GEARSET

Herringbone Gears Are Employed for Constant-Mesh and Third-Speed Trains

Internal Gears for Quiet Third Speed

All of the other four-speed transmissions now standard on American passenger-cars employ two pairs of internal gears to achieve whatever degree of quietness is attained in third speed. As I see it, the Warner Gear Co. and the Detroit Gear & Machine Co. designs (Figs. 3 and 4) are about equal in complexity and decidedly more complex than the Packard and the Pierce-Arrow spur-gear designs. Next in order and only slightly more complex is the Chrysler design (Fig. 5), followed by the early Durant[2] as the most complex of all.

Possibly a little less complex than any of these four internal-gear designs is the Johnson gearset, shown in Fig. 6 but not used as standard on any American passenger-car so far as I am aware. In this design, the tail-shaft is not co-axial with the clutch shaft. It is offset a distance equal to the eccentricity of the internal gears and their spur pinions, so that the drive never is direct but is through two pairs of internal-external gears in both third and fourth speeds. This is a disadvantage that some may regard as more than offsetting any gain in simplicity.

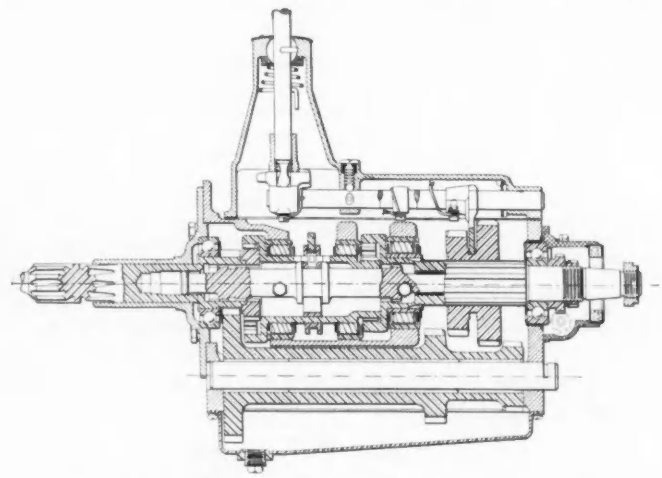


FIG. 3—WARNER GEAR CO. FOUR-SPEED GEARSET

This Design as Used in One Marmon Model and on Several Other Makes of Car Has Internal Gears and Two Reductions in the Constant-Mesh Train

Some of the seven four-speed gearsets referred to and illustrated herein have been described in papers presented before the Society. The drawings reveal most of the essential particulars regarding the others. It is worthy of note, however, that the internal-gear types require three or four more roller-bearings than do the spur-gear types, and, in addition, substitute for some of the spur gears internal gears cut on flanged or sleeve members that are much more expensive to make than ordinary spur gears and seemingly more expensive

than the herringbone type. Then, too, more extra parts, such as positive tooth-clutches, are required than in the spur-gear type.

In the Chrysler design the constant-mesh train involves three spur and one internal gear instead of two simple spur gears, as in all other designs of the internal-gear type except the Durant. While this complete train is loaded only in reverse, low and second speeds, none of which is much used, the extra gears turn constantly and cannot be expected to add to quietness of operation. The Chrysler gearset comprises a total of more than 100 parts and is far from being an easy unit to manufacture or assemble. For use in servicing it, no less than 13 special tools are recommended, and the instructions for servicing fill a 40-page booklet.

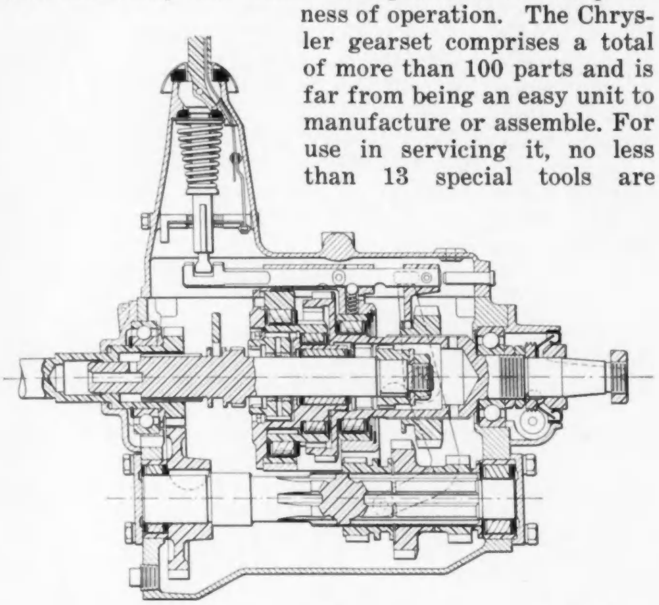


FIG. 4—DETROIT GEAR & MACHINE CO. FOUR-SPEED INTERNAL-GEARSET AS USED IN FRANKLIN CARS

Even more complex is the early Durant design. A drawing of a later and somewhat simpler Durant design is reproduced in Fig. 7. It, however, has more bearings than other makes. The earlier design has five roller and three ball-bearings for supporting the gears and other parts composing the main-shaft assembly alone. The secondary shaft is splined and carries a pair of shiftable gears as well as the keyed-on constant-mesh gear. It also drives a vane pump required for feeding lubricant to various parts. In this case the constant-mesh train alone involves three pairs of gears, so that the drive in low and second speeds is through four pairs of gears. All this is the price paid in efforts to obtain a quiet third speed. It is not difficult to see why engineers who have not been convinced of the superiority of four-speed gearsets say that the end does not justify the means.

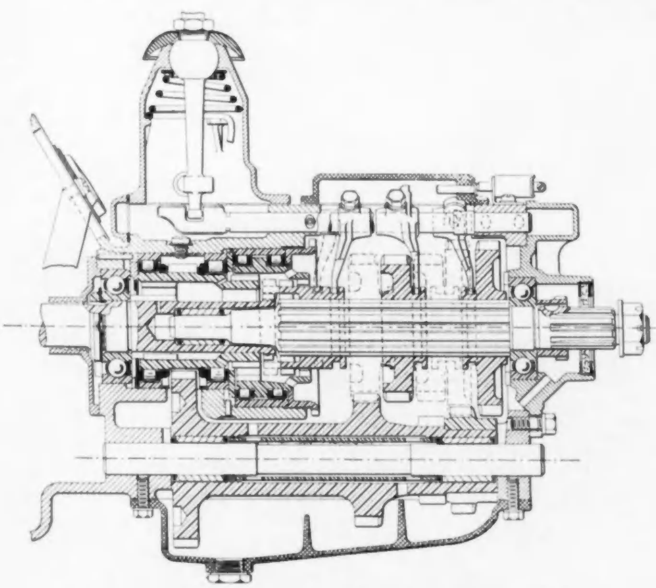


FIG. 5—CHRYSLER FOUR-SPEED INTERNAL-GEAR TRANSMISSION

This Design Has Two Reductions in the Constant-Mesh Train

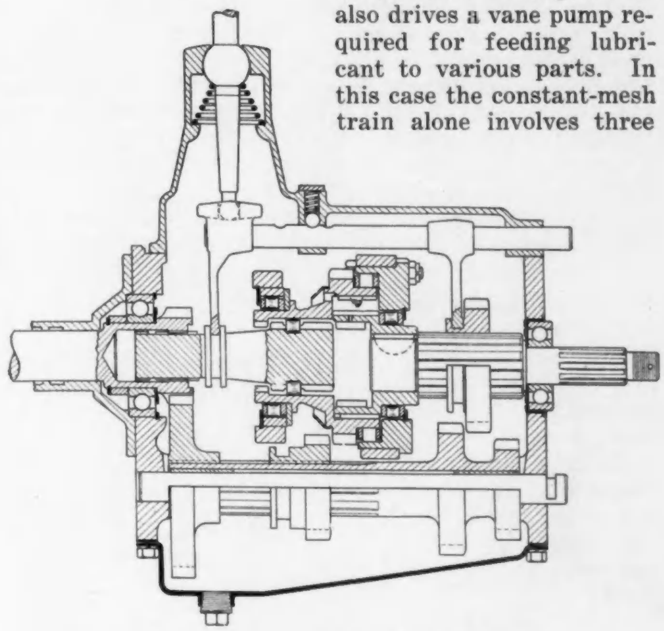


FIG. 6—JOHNSON FOUR-SPEED GEARSET

No Direct Drive Is Provided, as the Clutch Shaft and Tail-Shaft Are Not Co-Axial

Achievements of Three-Speed Advocates

In fairness to those who favor four-speeds, it is proper to inquire what the three-speed advocates have done toward achieving quieter operation, easier shifting and longer life of the car without undue sacrifice in general performance. Part of the answer has already been given in the discussion of rear-axle gear-ratios.

Both the Ford and Chevrolet cars have fairly large engines for the weight of the car and therefore have

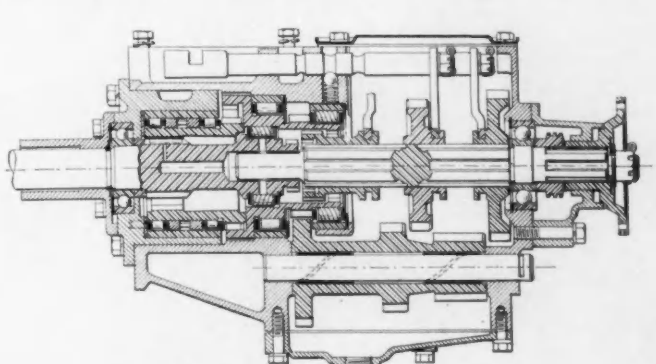


FIG. 7—LATEST DURANT FOUR-SPEED DESIGN

Three Reductions Are Provided in the Constant-Mesh Gear-Train

been able to use rather small reductions in rear-axle gears, thereby keeping the engine-speed low without sacrificing good general performance. That their gearsets are simple in design is apparent from Figs. 8 and 9. Ease in shifting and the degree of quietness in operation are not greatly different from those of other cars in the low-price range. The point to be emphasized in their case, however, is that high engine-speeds, which, as I see it, are largely responsible for bringing the four-speed gearset into the picture, are not essential if a sufficiently high power-weight ratio can be obtained in some other way.

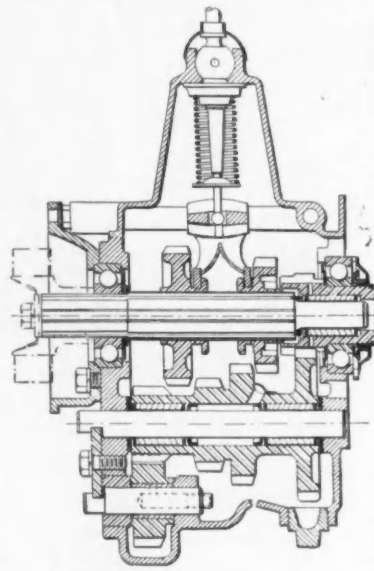


FIG. 8—MODEL A FORD THREE-SPEED SET

This Is a Good Example of a Design for Low Manufacturing Cost

More needs to be done to reduce car weight without going to high-engine speeds. Chassis are now made stiff and heavy to accommodate a body that is made heavy to keep it stiff, for present-day bodies do not stand much flexing without becoming noisy or failing in some other respect. Then a heavier and more powerful engine is required or its speed is boosted to keep the power-weight ratio high. The next step in this vicious cycle is a heavier and more complex gearset. If a start were made by building a body lighter and either redesigning it so that it could flex without injury or mounting it on the chassis frame which supports it so that road inequalities would not cause flexing, success in reversing the cycle might be attained. There is more in this suggestion than may appear on the surface. It is receiving serious thought in some circles and could be studied to advantage by every automobile engineer.

Herringbone Gears Used for Quietness

Most three-speed gearsets fall short on the score of quietness in second speed, and, to some extent, in respect to ease in shifting by the inexpert driver, but some steps to remedy those shortcomings have been taken. In fact, the Reo Motor Car Co.'s use of herringbone gears for constant-mesh and second-speed trains antedates their use by Pierce-Arrow, although in the Reo installation, shown in Fig. 10, no extra roller-bearing is employed for mounting the second-speed herringbone gear on the main shaft. This gear merely rides free on the main shaft until connected thereto by the positive clutch. This clutch is light and of small diameter and consequently makes shifting relatively easy. Both pairs of herringbone gears remain in constant mesh and run idle in all except the second-speed positions, which is not a serious disadvantage but one that well might be avoided if a simple way of doing so could be found. In other respects this gearset seems to meet every requirement. Except for the herringbone gears it should cost but little if any more than conventional types. Incidentally, it has a separately-cast bell-housing, a feature that is incorporated in many modern gearsets and that is designed to reduce manufacturing cost. Lead-base lubricants are recommended for use in this gearset.

Another three-speed gearset designed to combine quietness in intermediate gear and ease of shifting is the Warner Gear Co. transmission used for a time on the smaller Franklin chassis. This is of the internal-gear type and is a virtual replica of the Warner company's four-speed design except that one of the two spur gears on the primary shaft has been omitted and the unit shortened a corresponding amount. Although relative quietness and ease of shifting are said to be attained, the complexity of the design is not very much less than that of the company's four-speed unit.

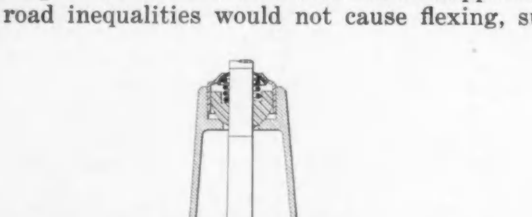


FIG. 9—CHEVROLET THREE-SPEED GEARSET

A Good Example of a Compact Conventional Design of Low Production Cost

FIG. 10—REO THREE-SPEED GEARSET

Herringbone Gears Are Used for Constant-Mesh and Second-Speed Gear-Trains for Quiet Operation

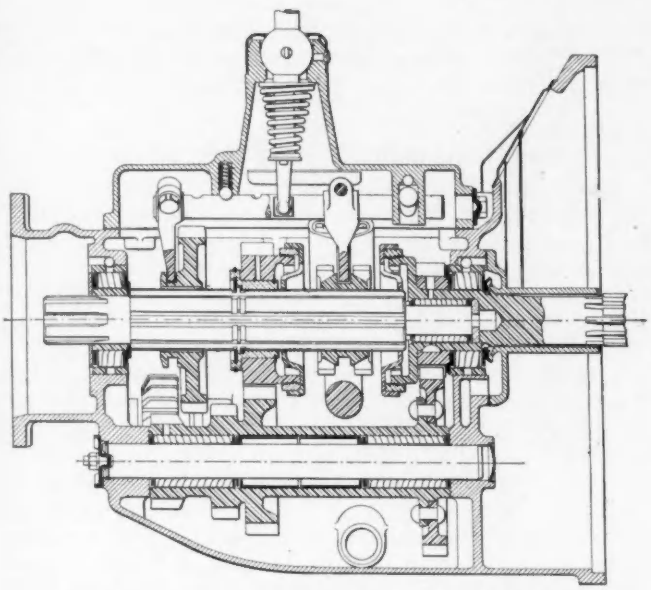


FIG. 11—CADILLAC SYNCHROMESH TRANSMISSION
Friction Clutches Are Used To Facilitate Shifting into Second and Top Speeds

Synchromesh Unit for Quiet Shifting

One other design in extensive use incorporates a device intended to facilitate shifting without gear-clashing. This is the Syncromesh unit employed in Cadillac and La Salle cars and illustrated in Fig. 11. It has only straight spur gears arranged in approximately the conventional way except that the intermediate-speed pair is in constant mesh. In consequence, the main-shaft gear of this pair is mounted loosely upon a bushing keyed to this shaft. When driving, this gear is connected to the shaft by a splined coupling having external teeth that mesh with internal teeth cut on the inner surface of a sleeve made integral with the gear. The external surface of this projecting sleeve forms the inner member of a cone clutch. The outer member of this clutch is a steel drum with bronze lining, the drum having inwardly projecting spokes that engage with the splined shaft.

When a shift into intermediate speed is to be made, the two conical surfaces first are brought into engagement, causing the shaft and the coupling which it carries to rotate at the same speed as the gear about to be engaged. Further motion of the coupling brings its teeth, some of which are cut away to enable the coupling to pass between the spokes of the drum, into engagement with those inside the gear projection, thereby positively locking the gear to the shaft. This latter motion causes no clashing, since the teeth to be engaged are turning at the same speed. A precisely similar arrangement of clutch faces, drum and coupling is used for the shift into direct drive. Thus it is virtually impossible for even a novice driver to clash gears in shifting from second to top speed or vice versa under almost any conditions.

Extremely easy shifting is accomplished, but the mechanism for bringing it about involves a considerable number of extra parts, including those of a rather

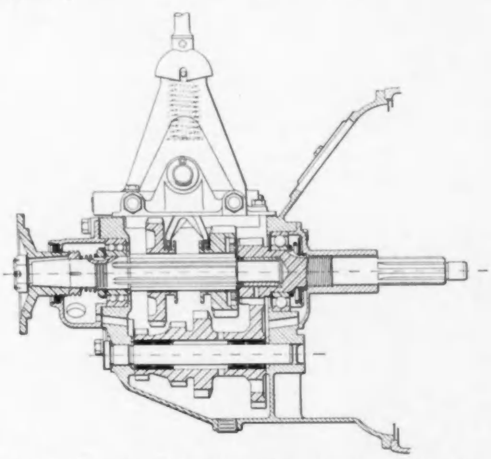


FIG. 12—STUDEBAKER DESIGN OF CONVENTIONAL THREE-SPEED GEARSET

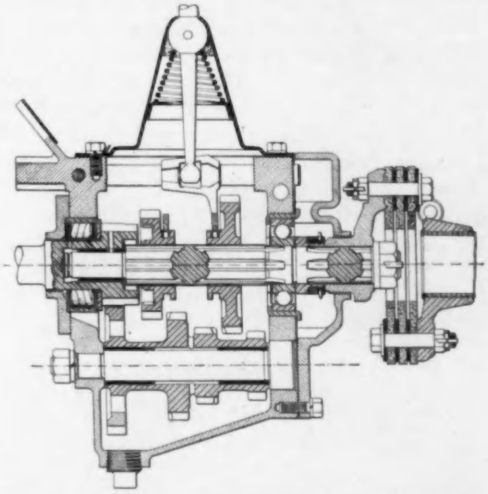


FIG. 14—NASH THREE-SPEED GEARSET
A Fabric-Disc Universal-Joint Tends To Reduce the "Telegraphing" of Vibration and Noise from Rear Axle to the Gearbox. One of the Main-Shaft Bearings Is a Straight-Roller Type

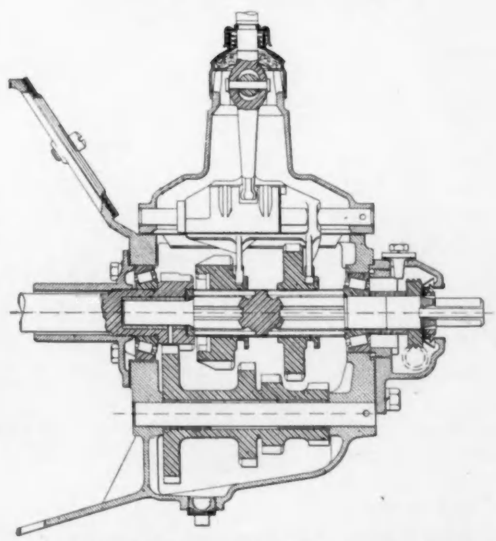


FIG. 13—WHIPPET THREE-SPEED DESIGN INCORPORATING TAPER ROLLER-BEARINGS ON THE MAIN SHAFT

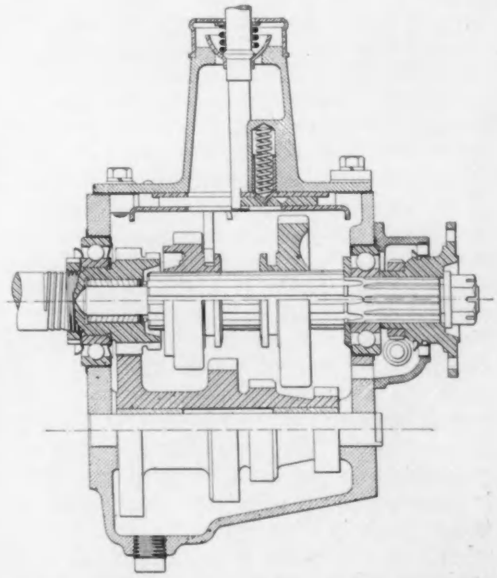


FIG. 15—MUNCIE DESIGN THAT IS TYPICAL OF GENERAL MOTORS THREE-SPEED-GEARSET PRACTICE

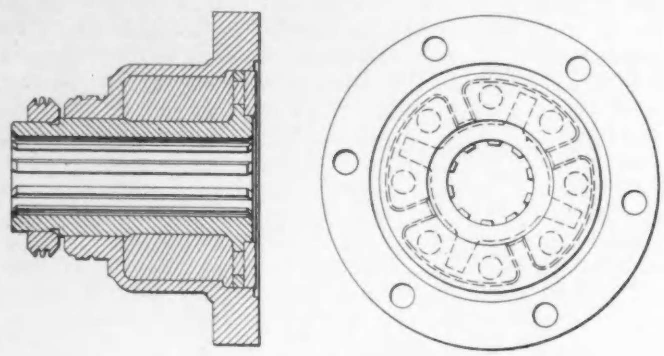


Fig. 16—Oakland Rubber-Cushioned Hub Assembly for Driving Propeller-Shaft

complex yoke arrangement which moves the clutch drums into engagement and then back into neutral position after the gears are engaged and before another shift can be made. The yoke mechanism includes a pair of oil plungers or dashpots arranged to automatically prevent excessively rapid shifting.

Aside from items already mentioned and the use of Hyatt roller-bearings on the main shaft, where ball-bearings ordinarily are employed, this gearset presents no noteworthy departures from conventional practice. Careful manufacture of the gearing is depended upon to assure the degree of quietness required. Reverse and low-speed gears are shifted in the usual way.

Means Available for Reducing Noise

Other gearsets used on American passenger-cars differ chiefly in details of construction. (See Figs. 12 to 15). The tendency has been toward compact designs, partly to limit weight and cost and partly to reduce the distance between bearing centers, thereby making the shafts stiffer, decreasing deflection and helping to reduce noise in operation. Other items tending to reduce noise deserve mention. One is the mounting of the gearcase on rubber supports which help to prevent vibration from being transmitted to frame and body. A second is the use of rubber in the drive between the gearset and axle, as in the rubber-hub coupling used on the tail-shaft of the Oakland gearset, shown in Fig. 16. This hub is said to be very effective in reducing high-speed gear rattle and the "telegraphing" of axle noise from the rear axle to the gearset. It also adds a desirable element of flexibility to the drive.

A further decrease in noise might be realized by mounting the gearset independently of the engine and using cushioning couplings between. This would add some simple parts but would facilitate access to the clutch and make easy the detaching of the gearset. Fortunately, this operation is not required very often, for in some cars it is necessary to detach and move back the rear axle and gearset to remove and replace clutch facings. Flexible-hub clutch-discs also help to prevent noise from reaching the gearset.

One source of noise originating within the gearset is constant-mesh gearing. The reverse idler sometimes is at fault in this respect. It should not be meshed when not carrying a load, and the same well might apply to other gears. This would not only help to reduce noise but would reduce a considerable power loss involved in churning the gear lubricant. It is very likely that some gear noise might be prevented from reaching the ears of those in the car if some sort of insulating or noise-deadening material were added to the gearcase and its cover. Body-panel drumming has been reduced by a similar expedient.

Standard Gearshift Should Be Retained

Many years were required for all American makers to adopt the present standard three-speed gearshift positions, and any tendency to depart from this standard is unfortunate. I believe that, if we are to have four-speed under-geared gearsets, the three top speeds should require the same shifting as for the three-speed type, the emergency low-speed being latched out or otherwise arranged to prevent accidental engagement. Any other arrangement is likely to confuse operators in emergencies and lead to many accidents otherwise avoidable.

Questions of type of gear, tooth form and pressure angles probably should be left to gear specialists, but it should not be forgotten that the possibilities on these scores have not been exhausted. Helical gears apparently present possibilities, in the way of decreased noise, that should be investigated. Such gears can be shifted endwise in much the same way as straight spur gears if a small helix angle is used. With such an angle, also, the end thrust is so small that no special provision for taking end thrust is needed and it can be made to nearly cancel out anyway by properly arranging the gears.

Perhaps ultimately we can develop engines having torque characteristics that will make gearsets unnecessary, or satisfactory torque-changing devices that will not require gear changing. Until then it is well to remember that modern gearsets are not incapable of improvement. On the other hand, their chief virtue is their simplicity and low cost. It may pay to sacrifice a part of this virtue for other advantages, but be-

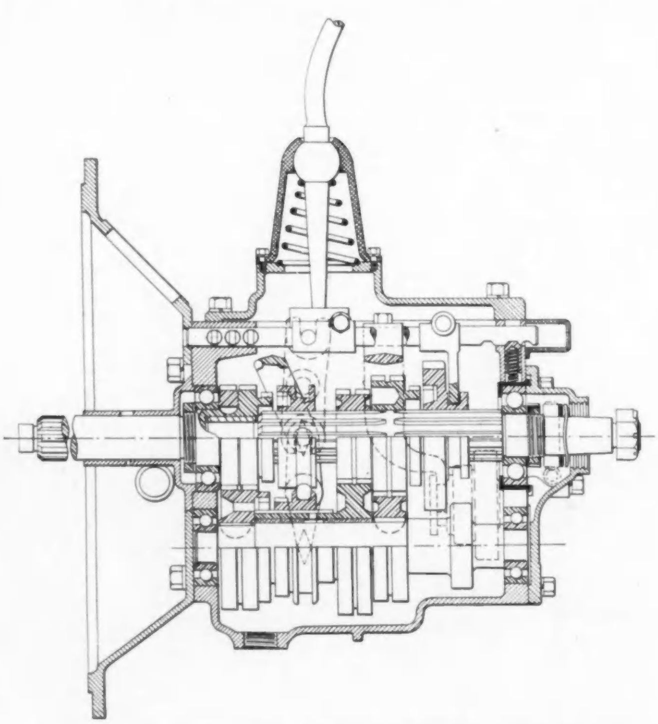


Fig. 17—Jones Four-Speed Over-Geared Design

Herringbone Gears Are Used for Constant-Mesh Second and Fourth Speeds. A Similar Design of the Same Make Drives Direct on Fourth Speed, and Another Has only Three Speeds

fore doing so the engineer should make sure that the gain more than offsets any losses involved.

SUPPLEMENT

Since the foregoing discussion of current gearset designs was prepared, particulars concerning other designs have become available and deserve comment.

The first of these is the Jones four-speed type having three sets of herringbone gears for the constant mesh, the second, and the over-drive fourth-speed pairs. Although these gears have a full 1-in. face, the gearset is the most compact of all the four-speed designs herein discussed. In the drawing reproduced in Fig. 17 an over-drive fourth speed is shown, but virtually the same design can be arranged for direct drive in fourth if desired.

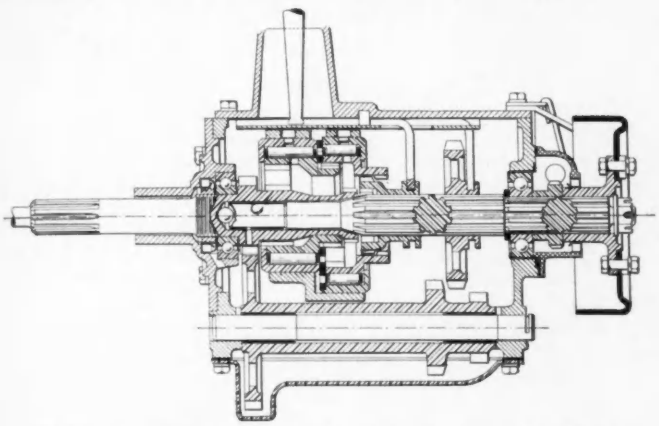


FIG. 18—MORSE CHAIN CO. THREE-SPEED GEARSET

This Design Has Internal-Gears. Similar Designs by the Same Company Have Four Speeds

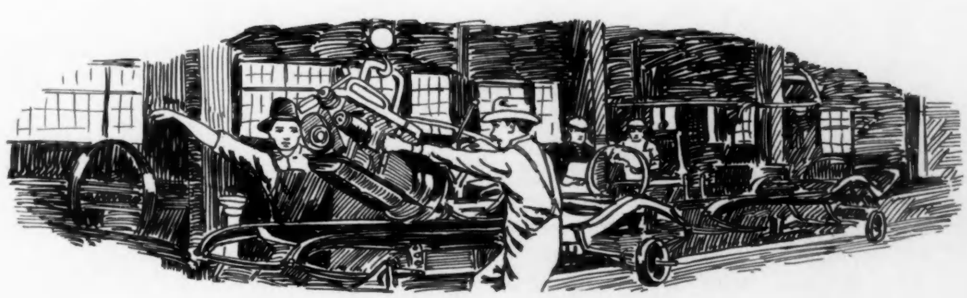
Both primary and secondary shafts are mounted on ball-bearings. The constant-mesh gear and second-speed pinion are keyed to the secondary shaft, but the fourth-speed gear is loose on this shaft and is engaged by a positive clutch keyed to a sleeve integral with this gear and meshing with an internally toothed sleeve cut integrally with the constant-mesh gear. The positive-clutch members for the second-speed gears are formed integrally with the first and second-speed gears on the primary shaft.

In all the positive clutches a coarse pitch and an even number of teeth are employed. Alternate teeth are cut about 3/32 in. short to facilitate engagement at high speed. Shifting is said to be noiseless and very easy at any speed. Quietness in operation is promoted not only by employing herringbone gears but by using teeth of different pitch in each ratio of gearing, the smaller pitches being used in gears that carry the lightest load. The pitch is said to be finer than in the usual conventional gearsets. Quiet operation in the over-geared fourth speed is aided also by pressing the fourth-speed pinion solidly onto the primary shaft and providing an extra-long bushing for the gear that meshes with this pinion. The same is true in third speed when the gearset is altered to drive direct in fourth. A very similar gearset having only three forward speeds is made by the same company.

Among the simpler designs of gearsets using internal-external gears for one ratio are those being developed by the Morse Chain Co. All four designs are very similar in major characteristics. The design illustrated in Fig. 18 has three speeds and drives direct in third. Another design drives direct on second and has an over-geared third speed. There are also two four-speed designs driving direct respectively on third and fourth, the first of these being over-geared on fourth speed.

Unlike some of the other designs using internal gears, the constant-mesh set involves only one reduction, being made in the conventional way with a pair of spur gears. In all four cases the secondary-gear shaft is fixed, with the gears turning upon it, as in most conventional designs. Only two roller-bearings are required for mounting the internal-external set, as against three in some other makes, but in the four-speed over-geared design an extra ball-bearing is employed on the primary shaft. Aside from these items and a slightly different arrangement of the internal gears, the design does not differ widely from other designs using internal gears.

In closing, I desire to express appreciation of the assistance given me by several members of the Society who furnished data, drawings and comments that were of considerable help in preparing this paper. Most of the tabulated statistics were taken from *Automotive Industries*.



Vapor-Locking Tendencies of Automotive Fuel-Systems

By W. C. Bauer[1]

Semi-Annual Meeting Paper — *Illustrated with* Photographs and Charts

Gas lock is stated to be caused by high temperature of the fuel and to be made manifest by failure of the engine to idle when hot after a fast run, uneven running during acceleration after idling, intermittent operation during a sustained high-speed run and, rarely, by complete stopping of the engine. Each of these manifestations has been isolated in the laboratory and its cause and location in the fuel system measured by temperature and analysis of the exhaust for carbon-monoxide and carbon-dioxide content. These findings are compared with data obtained in road runs at high atmospheric temperatures, six experimental fuels being used.

A heat differential of more than 5 or 6 deg. fahr. between the vacuum tank and the carbureter is shown to result in vapor lock. Design of the carbureter and fuel-feed system apparently has little if any effect on the temperature at which trouble occurs but does have a great effect on intensity and type of vapor lock.

The tendency to vapor lock places a very decided limit to the volatility of the fuel that can be marketed in the hot-weather season, as with higher volatility about one-fifth of the cars in use would have vapor-lock trouble. As a consequence, four-fifths of the cars are deprived of a fuel that is more satisfactory for them.

Too often in the designing of powerplants the effect of the hood, cowling and location of parts of the fuel system on operating temperature is disregarded. Cool carburetion is the surest safeguard against vapor lock. The ideal installation will deliver the fuel and air to the carbureter cool and have the necessary heat for vaporization applied to the manifold walls at a point between the carbureter and the cylinder.

Installations of internal-combustion engines in any service—motorcoach, tractor, automobile, truck, airplane or motorboat—can be, and are, made so that vapor locking never occurs under any conditions of hot operation. Other set-ups will gas-lock under normal warm-weather conditions. Hence, it seems probable that the trouble is more a factor of design than of fuel.

Gas lock is a condition caused by a high temperature of the fuel and is generally shown in four ways. The most common complaint is failure of the engine to idle after a fast, hot run or in traffic. Next in order of occurrence is intermittent or uneven running during acceleration after a period of idling. Intermittent operation or "locking" during a sustained high-speed run is also a form of this trouble. A rare but extremely annoying phase of vapor lock is a complete stopping of the engine.

In the laboratory we have been able to isolate and secure each of these manifestations and, by temperature measurement and carbon-monoxide and carbon-dioxide records, diagnose the cause and the location in the fuel system where the trouble occurred.

Stalling may be due to either a too lean or a too rich mixture for operation, depending on the type of the carbureter. With a carbureter of the plain-tube design having a separate idling tube and a jet feeding in above the throttle valve, the trouble is invariably a too lean mixture. Vapor bubbles form faster and faster in the idling riser as more and more heat flows back into the carbureter, until nothing but vapor is delivered to the idling jet and the engine stops.

In the case of an air-valve carbureter just the opposite occurs. The vapor bubbles, rising through a tube, the outlet of which is but slightly above the fuel level, act as gas lifts, pumping fuel out of the jet until a too rich mixture is supplied and the engine stalls.

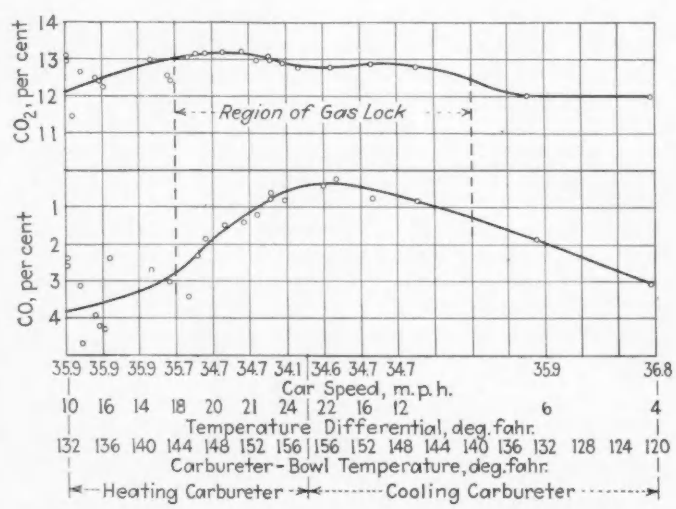


Fig. 1—Exhaust-Gas Analysis in Laboratory Tests

Variation in CO and CO_2 Content with Temperature Differential and Temperature of the Carbureter Bowl, Using Normal Fuel (Straight-Run Gasoline of 400-Deg. Fahr. End-Point) and Operating at No Load. When the Critical Vapor-Locking Temperature Is Reached, the CO Nearly Disappears and the Power Delivered Falls Off. Almost Complete Recovery to Normal Operation Occurs When the Carbureter Is Slightly Cooled

Uneven operation during acceleration usually marks the passage of vapor bubbles through the main jet, the cutting off of liquid flow being responsible for one or more of the cylinders receiving mixtures too lean to be

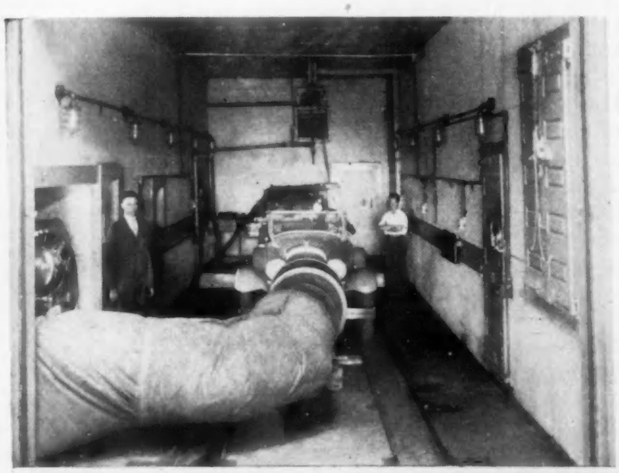
[1] M.S.A.E.—Research laboratories, Standard Oil Development Co., Elizabeth, N. J.

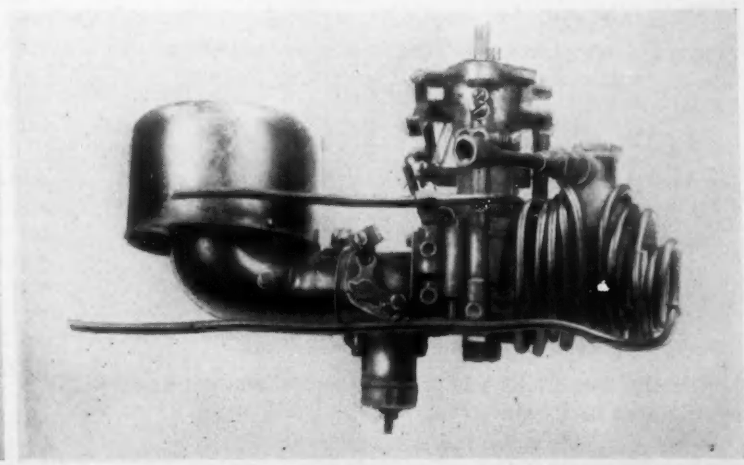
fired. With some plain-tube carbureters that function perfectly at normal temperatures a very decided lag or "flat spot" develops at the instant of throttle opening when operating at vapor-locking temperatures. This is due to the same gas-lift action that occurred in the air-valve carbureter. In this case the fuel that is pumped from the jet lies in the bottom of the air passage and gives a too rich mixture to be fired when the throttle is opened quickly.

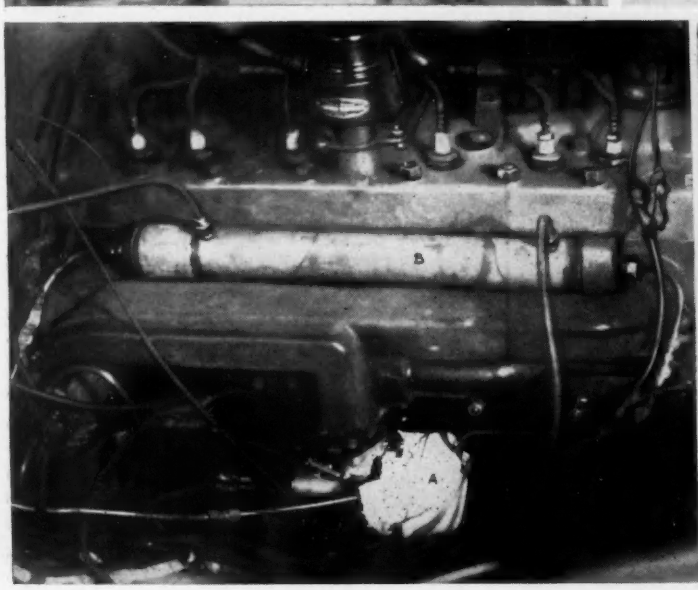
Gas Analyses Show Temperature Effect

By analyzing the exhaust gases, the CO-CO_2 recorder gives a very easily read picture of the sequence of action when gas lock occurs under continuous-operating conditions. Fig. 1 is a plot of such a record. CO and CO_2 are plotted against the temperature of the fuel in the carbureter bowl. There is a gradual improvement in combustion, shown by a higher content of CO_2 and a lower content of CO, until the critical vapor-locking temperature of the fuel is reached. At this point the CO nearly disappears and the horsepower delivered falls off considerably. The apparent drop in CO_2 is occasioned both by dilution of the exhaust sample with unburned charges coming through the engine and by the fact that the burning mixture is so lean that the peak of the CO_2 curve has been passed. There is a virtually complete recovery to normalcy of all readings when the carbureter is just slightly cooled.

Complete stoppage of the engine is caused by vapor lock in some part of the fuel system outside of the carbureter and may be due to a number of causes. The carbureter feed-line may pass too close to the exhaust pipe and thereby become so hot that the consequent formation of gas bubbles balances the liquid head and completely blocks the fuel-flow. Boiling in a gasoline filter-bowl will cause the same type of vapor lock. In







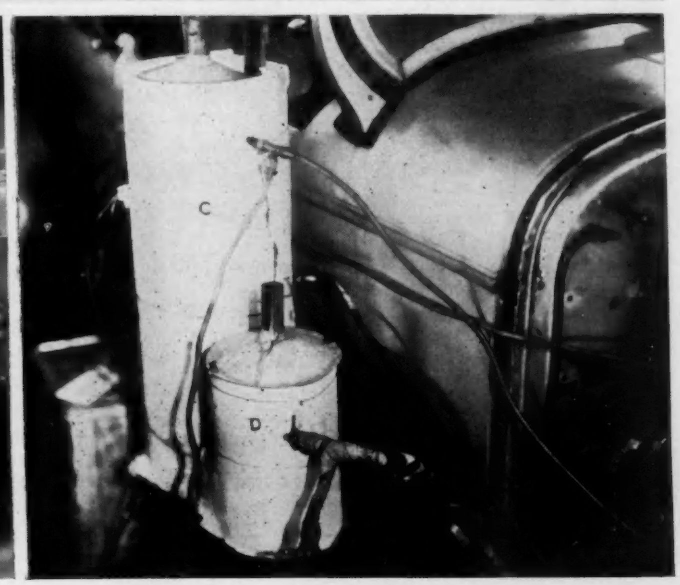


Fig. 2—(Upper Left)—General Arrangement of Test Car Anchored on Chassis Rollers, with a Wind-Tunnel Supplying Air to the Radiator

Fig. 3—(Upper Right)—Carbureter Fitted with Steam Coils for Maintaining a Predetermined Carbureter-Bowl Temperature; Asbestos Jacket Removed

Fig. 4—(Lower Left)—*A*, Carbureter Bowl Encased in Steam Coil and Heat Shield. *B*, Closed Steam-Apparatus for Preheating a Stream of Air To Impinge upon Carbureter Bowl, Used in Conjunction with *A*

Fig. 5—(Lower Right)—*C*, Vacuum Tank Removed from under the Hood and Encased in Steam Coil and Heat Shield To Maintain Predetermined Temperature of Fuel in Vacuum Tank. *D*, Heater for Heating Fuel between Vacuum Tank and Carbureter Bowl (Not Used)

SET-UP FOR MAKING VAPOR-LOCK TESTS UNDER CONDITIONS SIMULATING SUMMER ROAD-OPERATING CONDITIONS

every case in which complete stoppage of the engine occurred, the carbureter was found to be practically empty of fuel.

Almost all of the vapor-lock tests made by the research laboratories of the Standard Oil Development Co. were conducted on the chassis dynamometer. This dynamometer, being located within a controlled-temperature room, permitted runs to be made uninterruptedly at atmospheric temperatures of 95 to 100 deg. fahr. A comparative test was made to determine how closely operating temperatures obtained on the dynamometer checked those developed on the road. The comparative test consisted of a series of cycles of accelerating from a standstill to 40 m.p.h. in 1 min., maintaining 40 m.p.h. for 4 min., and then stopping and idling for 3 min. The temperatures, read at the end of the idling period, became approximately stabilized, as shown in Table 1. Except for the oil temperature, which ran considerably higher on the dynamometer, the correlation is very close.

TABLE 1—OPERATING TEMPERATURES IN ROAD-RUN AND CHASSIS-DYNAMOMETER VAPOR-LOCK TESTS

	Road Run, Deg. Fahr.	Chassis Dynamometer, Deg. Fahr.
Air in Front of Radiator	86	85
Fuel in Carbureter Bowl	146	148
Fuel in Vacuum Tank	139	141
Air to Carbureter	163	166
Intake Manifold	185	185
Water from Engine	172	175
Oil Sump	168	185

Six Experimental Fuels Used

For the establishment of the general principles governing gas lock, six experimental fuels were used, as follows:

(1) Normal, a straight-run gasoline of 400-deg. fahr. end-point
(2) Topped normal; No. 1 stripped to the pentane fraction
(3) Topped normal; 6.5 per cent butane
(4) Benzol blend; a naphtha of 390-deg. fahr. end-point with 30 per cent of commercial benzol added
(5) Topped benzol blend; No. 4 topped to the pentane fraction
(6) Topped benzol blend; 2 per cent butane.

To assure vapor lock with each of these fuels, the carbureter of the car chosen for further tests was heat-jacketed for temperature control. The vacuum tank was removed from under the hood, separately mounted and likewise heat-jacketed. Figs. 2 to 5 are photographs of the completed set-up. Runs were made both

TABLE 2—COMPARISON OF VAPOR-LOCK TEMPERATURES, IN DEGREES FAHRENHEIT

Fuel No.	Temperature in Carbureter Bowl at Time of Gas Lock	10-Per Cent Point, Engler Distillation	Liquid Initial Boiling-Point	Temperature at Which Vapor Pressure Equals 760 Mm. (29.921 In.) of Mercury
1	142	146	140	135
2	172	177	174	166
3	156	148	160	143
4	152	158	146	142
5	172	176	172	165
6	155	168	160	...

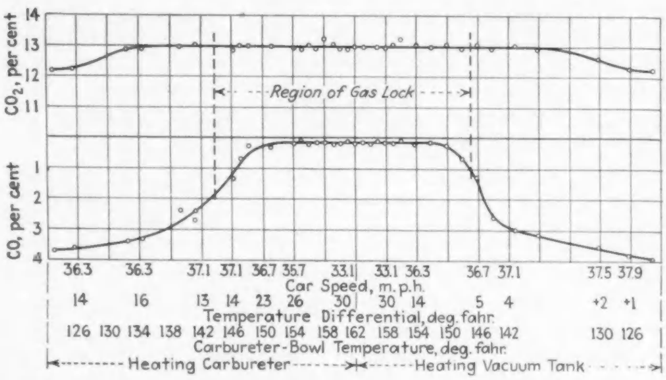


FIG. 6—EXHAUST-GAS ANALYSES UNDER CONDITIONS OF HEATING CARBURETER AND HEATING VACUUM TANK

Variation in CO and CO_2 Content with Temperature Differential and Carbureter-Bowl Temperature, Using Normal Fuel and Operating at Sustained Load at One-Quarter Throttle. Vapor Lock Was Established by Heating the Carbureter, and Normal Operation Restored by Heating the Vacuum Tank to within 5 or 6 Deg. Fahr. of the Temperature of the Carbureter Bowl

on the acceleration cycle before mentioned and under continuous touring-speed operation. The results obtained under both operating conditions were the same. Each fuel gave the first indication of gas lock when the temperature of the fuel in the carbureter bowl reached approximately the temperature of its liquid initial boiling-point. Table 2 gives the comparison of temperatures of vapor-lock indications with the temperatures of other characteristics that have been given as controlling vapor lock.

It is notable that, with a normal gasoline such as No. 1, there is a fair agreement between the locking temperature of the fuel, the 10-per cent off point on the distillation curve, the liquid initial boiling-point and the vapor-pressure balance temperature, but for the other fuels by far the closest correlation occurs when the locking temperatures and the liquid initial boiling-points are compared.

Small Temperature Differential Eliminates Trouble

During the tests with the heat-jacketed carbureter and vacuum tank a very interesting phenomenon was observed. When imperfect operation had been definitely established by heating the carbureter, heating the fuel in the vacuum tank to within 5 or 6 deg, fahr. of that in the carbureter bowl would cause the trouble to disappear. If this differential temperature was maintained it was possible to raise the fuel temperature of the carbureter bowl another 25 to 30 deg. fahr. without further difficulty being experienced. The CO-CO_2 picture of this condition shows that practically normal operation is obtained. Fig. 6 is a typical plot of the CO and CO_2 conditions under this operation. These curves show the establishment of vapor lock by heating the carbureter and restoration to normalcy by heating the vacuum tank.

This phenomenon explains why some vacuum-tank-equipped cars of a certain model have vapor lock and others of the same model do not. Just a little more heat in the vacuum tank or a little less in the carbureter would make the difference between a trouble-free installation and one that was almost inoperable under hot-temperature conditions, both using the same fuel.

Carbureter and fuel-feed design apparently had little or no effect on the temperature at which trouble oc-

curred but did have a great effect on the intensity and type of vapor lock. It was more difficult to maintain operation after locking had started with a plain-tube carbureter than with one of the air-valve type. On one particular car equipped with an air-valve carbureter the only indication of gas lock possible to obtain was a very distinct reduction of power delivered, extremely uneven idling and a distinct "flat spot" in operation when the throttle was opened quickly for acceleration.

It is these radically different manifestations of effect of vapor lock, the extremely varying intensities and the general instability of fuels having large percentages of light ends that make vapor-lock study so difficult.

Design Factors That Affect Vapor Lock

Vapor locking places a very decided limit to the volatility of the fuel that the refiner dares to place on the market during the season of his greatest sales volume. A lighter fuel would be easier to carburet and distribute uniformly in a multi-cylinder engine; it would be a cleaner-burning fuel, give better acceleration and have better antiknock properties. The automobiles on the road that would vaporlock if the liquid initial boiling-point of the fuel should be dropped 10 deg. fahr. would not exceed 20 per cent of the total. Yet for this 20 per cent the other 80 per cent are deprived of a fuel that would serve their requirements much more satisfactorily. We have in the last year tested for gas lock 21 cars and trucks, the product of 17 different factories. These 21 cars represent approximately 82 per cent of the Country's total production during this time, and the figures of 20 and 80 per cent are from these tests.

The factors of design that most affect vapor lock are the placing of the carbureter, the location of fuel-feed lines and fuel-pumps or vacuum tanks, the hood and mud-pan design, the position of the exhaust pipe and, finally, the air intake to the carbureter. Each of these details of design affects the temperature of the fuel in the carbureter and hence has a direct bearing on vapor lock.

There is another very inherent advantage of having a cool carbureter that seems to have been completely overlooked. If the induction system and compression ratio of the engine were designed to handle a cold-carbureted fuel, the temperature range through which the carbureter would have to function would be materially reduced. For instance, suppose that a carbureter is so placed that even under operating air-temperatures up to 100 deg. fahr. the carbureter never exceeds 125 deg. fahr.; then, for a temperature range from 0 to 100 deg. fahr. the operating range of the carbureter is 125 deg., while for a carbureter that heats to 160 deg., the range is more than 30 per cent greater. One popular car meets this qualification perfectly. Even at high-speed operation under heavy loads, carbureter fuel-temperatures remain below 130 deg. fahr. This same car is one of the easiest to start with small use of the choke at 0 deg. fahr., and its fuel economy is equal to that of any car.

To further emphasize the lack of standardization of operating conditions under which fuels have to function, Table 3 is presented. This gives the observed operating temperatures of all the cars tested to date.

TABLE 3—TYPICAL OPERATING-TEMPERATURES UNDER NORMAL FAST TOURING-SPEED CONDITIONS, IN DEGREES FAHRENHEIT

Car No.	Carbureter Bowl	Vacuum Tank or Filter Bowl	Air Temperature	Water Tank from Engine	Remarks
1	153	Vac. 144	95	180	Vapor lock
2	134	Vac. 133	99	188	No
3	167	Vac. 163	98	187	No
4	140	Vac. 115	98	182	Vapor lock, light fuels
5	128	Gravity Feed	101	201	No
6	118	F. B. 94	80	196	No
7	106	F. B. 95	82	185	Tendency
8	105	F. B. 95	80	168	No
9	148	Vac. 124	98	201	Tendency
10	130	F. B. 110	72	178	No
11	164	Vac. 97	80	168	No
12	126	Gravity Feed	80	...	No
13	137	F. B. 134	98–100	181	Tendency
14	164	F. B. 138	98–102	103	Vapor lock
15	153	F. B. 107	80	191	Tendency
16	152	F. B. 124	102	194	Tendency
17	133	F. B. 128	72	184	No
18	153	F. B. 137	95	204	Tendency
19	120	F. B. 108	82	180	No
20	140	Vac. 112	100	184	Vapor lock
21	132	F. B. 120	88	169	No

In studying the performance characteristics of some of the cars tested, the conclusion is inevitable that the development of the engine was conducted without thought of, or regard to, the effect of the hood, cowling, and so forth on the operating temperatures of the powerplant. It is not unusual to gain 3 to 5 per cent in horsepower delivery simply by raising the hood on the carbureter side of the engine. The ideal installation, from the viewpoint of both vapor lock and efficiency, is to place the engine and its necessary accessories in the car in such a way that engine-dynamometer conditions are approached. In other words, delivering both the air and the gasoline to the carbureter cool, with the necessary heat for vaporization put into the manifold above the carbureter, represents the best conditions of operation.

Shock-Absorbers

By J. M. Nickelsen[1]

Annual Meeting Paper — *Illustrated with* Charts

Experiments conducted to measure the movement of a car axle with respect to the car body when driving on the road, with and without shock absorbers fitted, are reviewed briefly and typical charts of the results are presented. These experiments are stated to have produced no information of benefit so far as riding-comfort is concerned, but did point the way to further study of the axle with respect to the car body.

Records taken from a car with its rear wheels driving on a pair of drums having wood blocks attached to their faces showed both axle and body movement with respect to the ground and indicated that the cycle of a shock absorber just after striking a bump would be that of the axle, after which it would gradually assume the cycle of the body. At a speed of 40 m.p.h., two-way hydraulic shock-absorbers function at axle speed and prevent the body being tossed about so much as it is without the shock-absorbers.

The graphic records indicate that the cycle of operation of the shock-absorber is almost entirely dependent upon axle movement rather than body travel or spring period.

Indicator diagrams showing the resistance of two-way shock-absorbers and the amplitude of movement of the arm at a speed of 100 cycles per minute are presented to show the type of control or resistance that can be secured and the variation in energy absorption with temperature change.

Discussers do not agree with all of the author's deductions and indicate that the subject will require much more study before an accurate means of measuring the action of shock-absorbers is developed that will enable the engineer to specify desirable functioning of the device in as definite terms as he now calls for engine performance.

Effect of change of temperature of the liquid in hydraulic shock-absorbers is debated at some length; synchronism of the car springs and seat cushions and the effect of shock-absorbers in reducing the amplitude of spring recoil are discussed, and one speaker states that he has developed a means for measuring the magnitude of the force transmitted between the axle and the shock-absorber arm at every instant of the cycle of operation. One discusser advocates a combination suspension of ring springs, which have excessive internal friction, and coil springs, which have too little internal friction.

Over the last few years the shock-absorber has gradually been gaining consideration from the car manufacturer until now virtually all cars have shock-absorbers of some type installed as standard equipment. The trend has been from the strap-type snubber to the one-way hydraulic shock-absorber, and now greatest interest seems to center in the two-way hydraulic device.

On account of the present interest in the double-acting shock-absorber, this paper will deal primarily with that phase of the subject. Before entering into the discussion of two-way shock-absorbers, I should like to picture clearly the movement of the shock-absorber and the zone that it is working in when it is installed on the car.

Some of the early experiments conducted in this connection were to measure the axle movement with respect to the car body on the road. This was done in the hope of securing some measure of the riding-qualities of an automobile. The results were of little or no value from this standpoint but mention is made of it in an attempt to point out the relative movement that occurs between axle and car body and the time interval of this action.

Fig. 1 is a graph taken over a gravel road at 25 m.p.h. with no shock-absorbers on the car. Curve *A* shows relative movement of the rear axle with respect to the car body. Curve *B* gives the time interval, as each break in the line indicates 1 sec. of time and was recorded by means of a chronograph. Fig. 2 is a similar graph over the same piece of road, conditions being the same except for the addition of two-way hydraulic shock-absorbers adjusted for average car-control.

Comparison of these two graphs shows that the phase relation of the axle with respect to the car does not change materially with the addition of shock-absorbers. A great many other comparative runs were made, all of which brought out the same general results as shown in Figs. 1 and 2. To derive from these curves anything of benefit, as far as determining riding-comfort was concerned, was impossible; however, they did point the way to a further study of the conditions under which the shock-absorber must function.

Movements with Respect to Ground

Hoping to get a clearer picture of the foregoing items as well as the movement of the car body with respect to the ground, a car was mounted with the rear wheels on a pair of chassis-testing drums. Wood blocks of various heights were placed on the drums and the rear wheels were driven by means of the engine. Curves were recorded showing both axle and body movement with respect to the ground.

A record taken of a car with the rear wheels on the drums is presented in Fig. 3. Blocks 1¾ in. high were used for bumps. The speedometer reading was 10 m.p.h., and no shock-absorbers were on the car. Curve *A* shows the axle movement with respect to the ground, and curve *B* the body movement with respect to the ground. Curve *C* is derived from *A* and *B* and gives the relative movement of the axle with respect to the car body. This Curve *C* is comparative with Curves *A* in Figs. 1 and 2 as taken on the road. When Curve *C* falls above

[1] M.S.A.E.—Associate professor, University of Michigan, Ann Arbor, Mich.

the horizontal line the springs are compressed more than their normal amount, and when it falls below the line they are compressed less than normally. As Curve *C* travels upward, a shock-absorber would be functioning in the compression zone, and when it travels downward would be working in the rebound zone. Curve *D* gives the time record as taken by the chronograph, each break in the line being 1/5 sec. This indicates an axle period of about 500 cycles per minute and a body period of about 80 cycles per minute. It is to be noted that if a shock-absorber were installed its cycle of operation just after striking the bump would be that of the axle and then it would gradually assume the cycle of the body movement as the axle movement dampens.

Fig. 4 is a comparative set of curves taken under the same conditions except for the addition of two-way hydraulic shock-absorbers with an adjustment giving very little resistance. Both the body and axle curves indicate less movement and quicker dampening. Curves *A*, *B* and *C* in Fig. 5 show the axle and body movement and the relative movement of the axle to the body taken under the same conditions as in Fig. 3 except for the speedometer reading, which was 40 instead of 10 m.p.h.

Fig. 6 compares with Fig. 5 except for the addition of two-way hydraulic shock-absorbers with the same adjustment as used in Fig. 4. At this speed the axle movement does not have time to dampen out either with or without shock-absorbers, but the car body is not tossed around nearly as much with shock-absorbers, which function at axle speed throughout, as the axle does not have time to dampen out before the wheel strikes the bump on the next revolution of the drums. That the cycle of operation of the shock-absorber is almost entirely dependent upon axle movement rather than body travel or spring period is very apparent from the foregoing records.

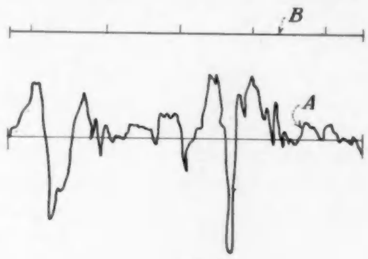


FIG. 1

Fig. 1—Taken on Gravel Road at 25 M.P.H. with No Shock - Absorbers. Curve *A* Shows Movement of Axle with Respect to Body. Curve *B* Gives Time Intervals in Seconds

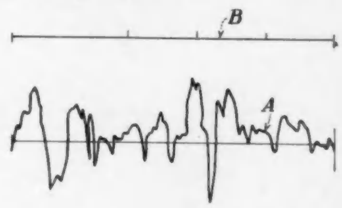


FIG. 2

Fig. 2—Taken under Same Conditions as Those in Fig. 1, Except for Addition of Two-Way Hydraulic Shock-Absorbers Adjusted for Average Car-Control

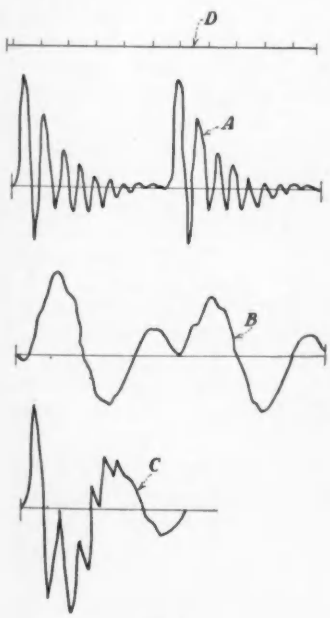


FIG. 3

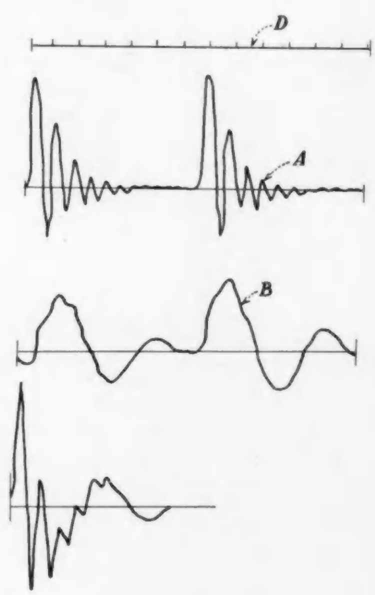


FIG. 4

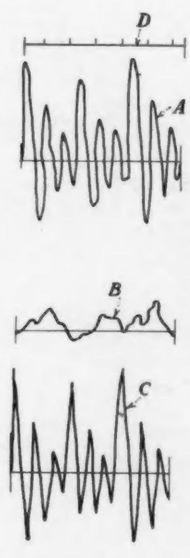


FIG. 5

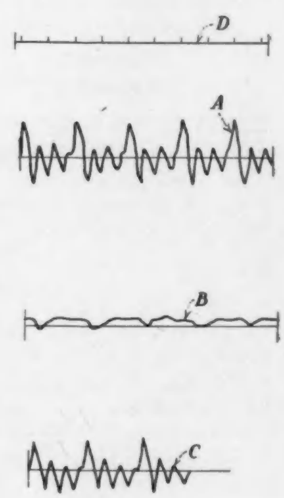


FIG. 6

OSCILLOGRAPH RECORDS OF CAR AXLE AND BODY MOVEMENTS

Fig. 3—Record of a Car with Rear Wheels Driven on Chassis-Testing Drums Fitted with Wood Blocks 1¾-In. High. No Shock-Absorbers Fitted; Speedometer Reading, 10 M.P.H. Curve *A* Shows Axle Movement with Respect to Ground; Curve *B*, Body Movement with Respect to Ground; Curve *C* Is Derived from *A* and *B* and Gives Relative Movement of Axle with Respect to Body and Is Comparative with Curve *A* in Figs. 1 and 2

Fig. 4—Comparative Curves Taken under Same Conditions as in Fig 3 Except for Addition of Two-Way Hydraulic Shock-Absorbers Adjusted to Give very Little Resistance. Curves *A*, *B* and *C* Correspond to Similarly Designated Curves in Fig. 3. Curve *D* in Figs. 3 to 6 Give Time Intervals in 1/5 Sec.

Fig. 5—Records Taken under Same Conditions as Those in Fig. 3 Except for Speedometer Reading, Which Was 40 Instead of 10 M.P.H.

Fig. 6—These Curves Are Comparative with Those in Fig. 5 Except for the Addition of Two-Way Hydraulic Shock-Absorbers with Same Ajustment as Used in Fig. 4

Desirable Amount of Shock-Absorber Resistance

Automotive engineers are fairly well agreed at present, I believe, that of the devices now on the market the two-way hydraulic shock-absorber gives the most desirable control of the car body, which is the factor of importance. Considerable difference of opinion seems to exist as to how much control the car should have as well as the way in which this energy should be taken up by the shock-absorber, by which I mean the amount of resistance the shock-absorber should have when the springs are compressing and the amount it should have on rebound. In general, I believe it is undesirable to have more than 15 or 20 per cent of the resistance on the compression side, with the added factor that the shock-absorber should not build into this resistance too fast. On the rebound side I think it is likewise desirable not to have too quick a build-up in pressure. As to free center-action, or movement of the axle before resistance is built up in the shock-absorber, in general it is desirable to have quicker control on the rebound than on the compression side. A factor which enters here is whether the absorber is placed on the front or the rear of the car. Often it is desirable to have the absorber on the front of a car build up its resistance very fast so as to steady the car at high speeds.

Figs. 7 to 10 are indicator diagrams of action of two-way shock-absorbers taken at a speed of 100 oscillations per minute. The vertical height represents pounds of resistance and the horizontal distance represents the displacement or movement of the shock-absorber arm. Above the horizontal line indicates the resistance on rebound, and below this line indicates the resistance on compression. Fig. 7 indicates a rather slow, gradual build-up with no resistance on compression. Fig. 8 shows a rather quick build-up on rebound, some free center on compression, and then a build-up to about 30 per cent of the rebound resistance. Fig. 9 brings out a gradual build-up on rebound and then maintenance of a uniform pressure, while on compression there is a long free center with a gradual build-up into high resistance at the end. Fig. 10 has the same general characteristics as Fig. 9 on rebound but differs on compression, in that it has less free center with a gradual build-up into resistance, which is about 10 per cent of the rebound side.

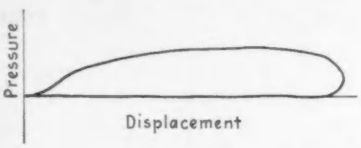


FIG.7.

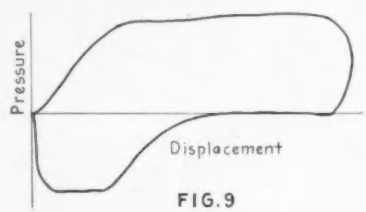


FIG.9

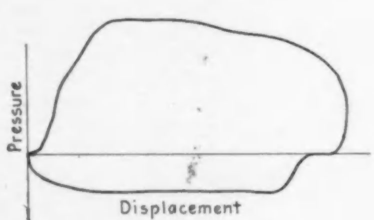


FIG.8

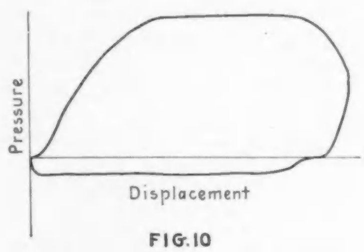


FIG.10

INDICATOR DIAGRAMS OF ACTION OF TWO-WAY HYDRAULIC SHOCK-ABSORBERS AT A RATE OF 100 OSCILLATIONS PER MINUTE

Vertical Height Represents Pounds of Resistance; Horizontal Distance Represents Displacement or Movements of the Shock-Absorber Arm. Above the Horizontal Line Indicates Resistance on Rebound; Below the Line Indicates Resistance on Compression

Fig. 7—Gradual Build-up with No Resistance on Compression

Fig. 8—Rather Quick Build-up on Rebound, Some Free Center on Compression and Build-up to about 30 Per Cent of Rebound Resistance

Fig. 9—Gradual Build-up on Rebound; Maintenance of a Uniform Pressure, Long Free Centre on Compression and Gradual Build-up into High Resistance at the End

Fig. 10—Same General Characteristics on Rebound as in Fig. 9 but Less Free Center on Compression and Gradual Build-up into Resistance, Which Is about 10 Per Cent of the Rebound Side

Energy Capacity Varies with Temperature

As the shock-absorber functions under the car, it absorbs energy that can be measured in some suitable unit. After the correct adjustment is once made on a car, it is desirable that this energy taken up by the shock-absorber remain a constant, or otherwise the car control will vary. The item that causes most variation in the energy capacity of a hydraulic shock-absorber with a definite setting is the change in oil viscosity with change of temperature. This energy variation in inch-pounds over a temperature change from 32 to 120 deg. fahr. is shown in Fig. 11. These curves were taken from standard makes of shock-absorbers just as they were removed from cars. Curves *A* and *B* are for shock-absorbers having a pressure relief and a metered orifice, while Curves *C* and *D* are for a metered orifice only. These curves show that the shock-absorbers with pressure relief have much less drop in capacity owing to this temperature change than have the units with only the metered orifice.

On account of the variation in the settings and control of these units, the results have also been plotted in terms of variation in capacity based upon 32 deg. fahr. as 100 per cent so as to give a better picture of the effect of this temperature variation. Fig. 12 gives the results plotted in this manner. The four curves are for the same respective shock-absorbers as the curves in Fig. 11. Shock-absorbers with the pressure relief, represented by Curves *A* and *B*, show the least drop in per cent capacity over this temperature range, since they retain 67 and 65 per cent respectively of their original capacity. The units with only the metered orifice, represented by Curves *C* and *D*, show considerably more drop in control, as *C* retains only 31 per cent of its initial capacity and *D* only 14 per cent.

I have attempted in this discussion to present only a few of the facts which appear to be most pertinent in connection with shock-absorbers and their action on the car, the type of control that can be secured through the use of the two-way hydraulic shock-absorber, and the possible variation in energy absorption with temperature change which occurs in some of the units on the market.

THE DISCUSSION

TORE FRANZEN[2]:—The shock-absorber art is as yet very young. There is plenty of room for individual opinions, and it will no doubt be years before we, as engineers, can specify desirable shock-absorber functioning in as accurate terms as we today call for engine performance, flow and pressure in pumps, output of generators, storage-battery capacity and so forth. Any technical contribution that will help us to design shock-absorbers with definite and desirable functioning is therefore very welcome and valuable.

Professor Nickelsen's paper is valuable in this respect. It shows us how a shock-absorber has to function at speeds and amplitudes which have in the past been considered important, yet not of such importance that we have the abundance of test information we now possess for shock-absorbers operating at higher amplitudes and lower velocity.

The present conventional method of dynamic testing of shock-absorbers is with relatively high amplitudes and the low speeds seldom above 120 cycles per minute. In the same instrument some relationship undoubtedly exists between the distinct functions referred to, and the present routine dynamic testing can be continued as standard inspection procedure, whereas the laboratory equipment should be revised to measure all possible amplitudes and speeds. This will enable the engineers to intelligently distinguish complete shock-absorber reactions.

In hydraulic instruments having no relief valves or metering devices, the resistance should go up with the square of the velocity. My belief is that this is not entirely true, as the walls of such a shock-absorber will undoubtedly deflect and give a wider bypass than that established by the metered orifice.

[2] M.S.A.E.—Research engineer, Chrysler Corp. Engineering Laboratories, Highland Park, Mich.

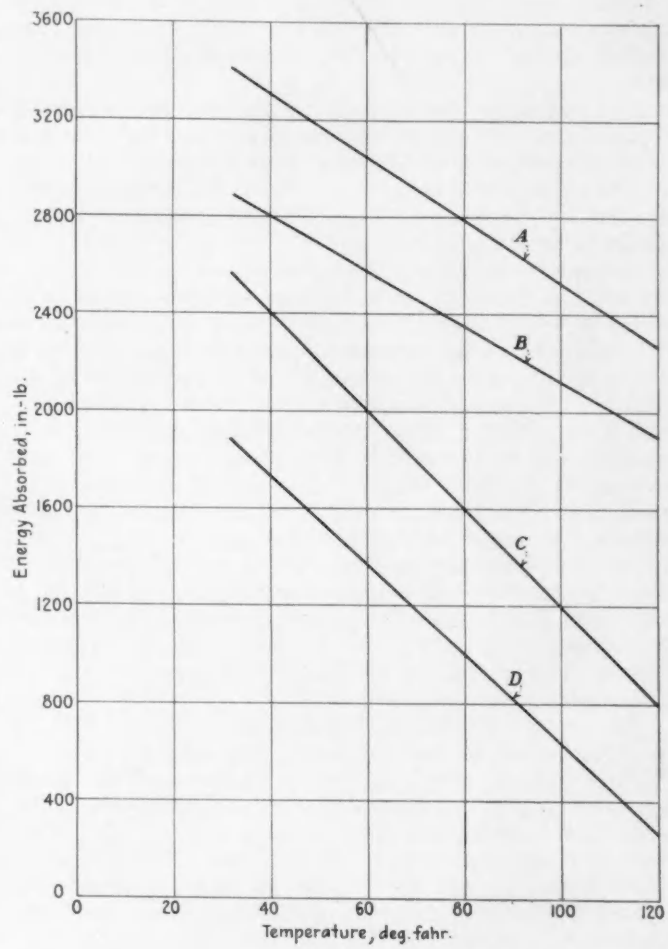


FIG. 11—VARIATION IN ENERGY ABSORPTION OF A HYDRAULIC SHOCK-ABSORBER WITH CHANGE IN OIL VISCOSITY WITH TEMPERATURE CHANGE FROM 32 TO 120 DEG. FAHR.

These Curves Were Taken from Standard Makes of Shock-Absorbers Just as They Were Removed from Cars. Curves *A* and *B* Are for Absorbers Having a Pressure Relief and a Metered Orifice; Curves *C* and *D* Are for a Metered Orifice Only

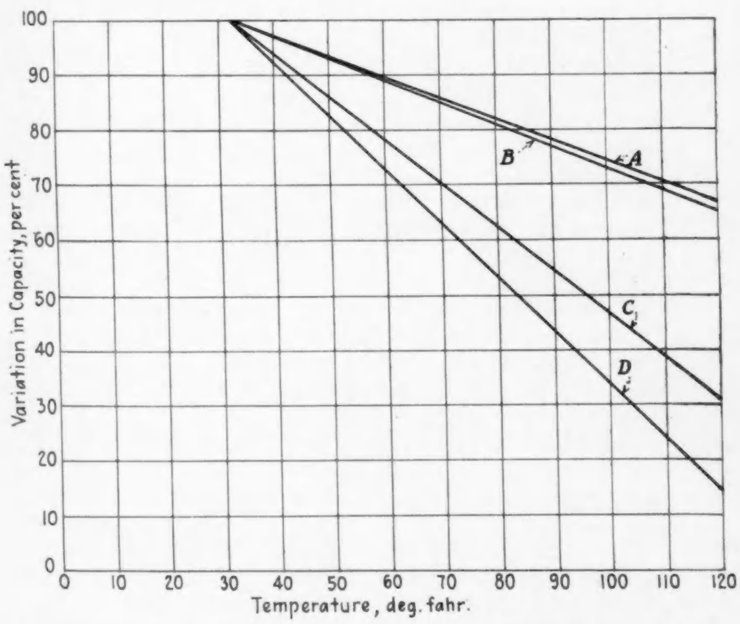


FIG. 12—RESULTS SHOWN IN FIG. 11 PLOTTED IN TERMS OF VARIATION IN CAPACITY BASED ON 32 DEG. FAHR. AS 100 PER CENT

The Four Curves Are for the Same Respective Shock-Absorbers as the Curves in Fig. 11. They Give a Better Picture of the Effect of Temperature Change than the Curves in Fig. 11, on Account of Variation in Setting and Control of the Several Shock-Absorbers

Axle Periodicity Increased Greatly at Speed

In Figs. 3 and 6 Professor Nickelsen shows how the axle periodicity is increased by the application of shock-absorbers. The short records presented in the paper indicate that the periodicity is increased a relatively small amount at low car-speeds, whereas a decided increase occurs at higher car-speeds. This increase in periodicity is likely to follow a curve that bears a definite relationship to car speeds. The efficiency of the hydraulic-type shock-absorber is perhaps limited to the proportion in which it will increase frequency and axle vibrations. The fact that the several attempts to use hydraulic shock-absorbers on high-speed racing cars were unsuccessful, in some degree proves this theory. Some of the failures to control fast vehicles were due to overheating. Friction two-way shock-absorbers have so far been the only successful damping instrument for

very high-speed vehicles. This may be owing to the fact that a slight decrease in frictional resistance occurs with increased velocity. In view of this, there may be room for development of such hydraulic shock-absorber orifice and valve as have definite relationship to speed of vibration.

I cannot subscribe entirely to the professor's opinion of division of control of compression on the rebound side. My experience indicates that a resistance on the compression side exceeding 10 per cent of the rebound control at a speed of 110 cycles per minute gives undesirable results.

When the author states: "On the rebound side, I think it is likewise desirable not to have too quick a build-up in pressure," he will not have a unanimous following, as some instruments are actually being built today with nearly instantaneous build-up to maximum control. Personally, I agree with him, as I have experienced, in working with shock-absorbers with combined orifice and blow-off control, that a highly restricted orifice which causes immediate valve-lifting and maximum control gives unpleasant results, particularly at low car-speeds. My experience has been that free center-action, of the type indicated in Fig. 9, gives an unpleasant ride. Such shock-absorber action is also likely to set up a vibration that will cause fenders, headlights and radiator to shake in an otherwise steady car, which phenomenon is perhaps due to the fact that checking occurs at maximum axle-velocity.

Professor Nickelsen also indicates the desirability of a different action in front and rear shock-absorbers. This is undoubtedly true, and the day may come when we shall specify entirely different shock-absorbers for front-end installation. The curves given in Fig. 11 are not quite clear, and I shall ask the author how they were arrived at. In our laboratory we have recently conducted a number of experiments with shock-absorbers at zero temperature which indicate that the difference in efficiency between 70 and 0 deg. fahr. is greater than that shown by Professor Nickelsen between 120 and 32 deg. This fact leads us to believe that somewhere below 32 deg. a distinct offset occurs in the otherwise straight curve shown in Fig. 12.

Oil-Viscosity Variation Uncertain

PROF. J. M. NICKELSEN:—Relative to the control on compression compared with rebound, I believe, in general, that 10 per cent is about as high as can be used. On the other hand, in some cases it seems to me that more control can be used safely for the front end of an automobile.

As to the temperature-variation condition, in a study of oil viscosity which I made some time ago, my guess before running any tests was that the viscosity curve should have a characteristic on the order of the Curve in Fig. 13. If the abscissa represents temperature and the ordinate represents Saybolt viscosity, then the curve takes on the general form of the curve shown for virtually all liquids used in shock-absorbers.

My study along that line was not 100-per cent complete. I carried it experimentally, however, from a temperature around 120 deg., as I recall my high point, down to about 5 deg. below zero, and, with the exception of kerosene and alcohol, the lines all took on the general form of this curve. Those liquids, of course, had lines that were virtually straight, with a very slight up-slope through that temperature range. If kerosene and alcohol could be used in shock-absorbers, they would be what we might term ideal liquids. We should then expect a straight line instead of a curve as indicated in Fig. 13. The reason for drawing those straight lines is that not a complete enough study has been made to satisfy me that that curve does take on that form. Of the shock-absorbers tested, the general appearance of the curve seems to be a straight line over the temperature range in which we have worked. I intend to go on with the tests down to lower temperatures, to see if I can establish a definite curve. Some of the shock-absorbers seem to have a more or less convex curve instead of the concave curve as we should expect. I do not know why they should do that and consequently am merely taking my two end-points and drawing a curve through them to represent the mean of conditions.

Relationship of Resistance to Viscosity

CHAIRMAN W. R. GRISWOLD[3]:—In connection with shock-absorber liquids, some work done on various types of paraffin-base lubricants of varying degrees of viscosity showed that when these viscosities are plotted on logarithmic paper they show a straight line, which indicates that they have the curve shown in Fig. 13, and that the viscosity tends to go very rapidly toward infinity as the temperature gets down very low, around zero or 5 deg. below zero. These tests were made by forcing the liquids through an orifice at about 300-lb. pressure so that the viscosity below the ordinary pouring temperatures could be studied.

I have noticed that in shock-absorbers the expected ratio of resistance for cold and for hot conditions does not obtain as indicated by these viscosity curves, as between high and low-viscosity liquids. Some of the results of tests seem to be contradictory. I think that indicates that possibly some mechanism operates in the lubricant that passes through the orifice. I do not know what it is, but possibly a great temperature change occurring as the liquid passes through the orifice has something to do with it. All the work that most of us have done on shock-absorbers in trying to correlate temperature with resistance has been on the measuring of the temperature of the main body of the liquid rather than the temperature that is causing the real resistance. Some of the inconsistencies seem to point in that direction.

R. E. WILKIN[4]:—The resistance to flow of fluids in shock-absorbers may not be in direct relation to their Saybolt viscosities. In fact, the viscosity of alcohol or kerosene cannot be determined in the Saybolt instrument, since it is not accurate on any fluids showing a value below 40 sec.

The resistance to flow can be determined from the absolute viscosity of the fluid, and this viscosity can be calculated if its Saybolt is above 40 sec. and its specific gravity at the operating temperature is known.

Such low-viscosity fluids as alcohol and kerosene do change in viscosity with temperature, although not as rapidly as higher-viscosity oils or glycerine.

Another point to be considered in studying the performance of shock-absorbers at low temperature is the effect of cold test. This may change the apparent viscosity of the oils at temperatures of 30 to 40 deg. above their A.S.T.M. cold-test.

[3] M.S.A.E.—Engineer in charge of design analysis, Packard Motor Car Co., Detroit.

[4] M.S.A.E.—Automotive engineer, Standard Oil Co. (Indiana), Whiting, Ind.

Use Low-Range Springs and Reduce Amplitude

L. K. SNELL[5]:—About 10 or 12 years ago Walter Keys conceived the idea of recording the action between the passenger-car body and the action of the accessories. He and I attached lights to the hubs, the fenders above the front and rear wheels, and to the passenger's neck. We prevailed upon the company to build a road in which we could put bumps or holes or anything we wanted; otherwise it was level and smooth.

We made a great many pictures at night, setting a camera up at a measured distance from the track so that the lines traced by the light on the plate were measured in hundredths of an inch. These gave an absolute record of the relationship of all parts of the body and wheels and the action in all positions of the car on the road. We tried to find some basic principles by which riding-qualities could be governed. The first thing we learned was that the period of recoil of the springs is the ratio of the suspended load to the rate per inch of the springs, and that the shock-absorber did not affect this period appreciably. I believe that Professor Nickelsen's charts will verify that.

So it seems to me that we must use a low-rate spring to lower the period of recoil, and use a shock-absorber to reduce the amplitude of that recoil; in other words, do some work in something besides the spring that will not pay the energy back into the body. The period of recoil of the seat cushion in relation to the spring is just as important in riding-qualities as the car spring itself. The reason for two-way shock-absorbers, as far as I can see, is that a spring of a very much lower rate and having a lower period of recoil can be used and some of the work can be done in the shock-absorber, which will not pay it back into the body in the form of velocity. Thus the amplitude of the recoil can be reduced.

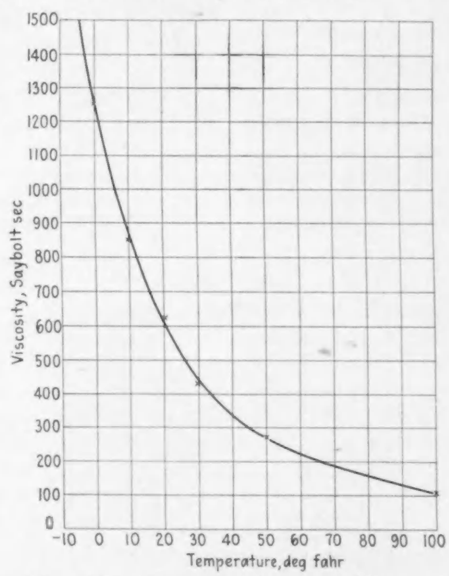


FIG. 13—ASSUMED GENERAL FORM OF VISCOSITY CURVE OF SHOCK-ABSORBER FLUID WITH VARIATION OF TEMPERATURE

Experimental Tests Did Not Include Kerosene and Alcohol, Which Would Be Expected To Give a Straight Line Instead of a Curve and Might Therefore Be Termed Ideal Liquids if They Could Be Used

The synchronism between the period of recoil of the body, the car spring and the cushion is of greatest importance, and the synchronism between the front and rear springs is also important.

One of the unfortunate conditions is that we have to take from the makers spring steel that has too much variation in the thickness. As the flexibility of the spring varies with the cube of its thickness, it is a hard problem to get a uniform rate per inch in all the cars that are built.

Means for Measuring Energy Dissipation

E. W. WEAVER[6]:—I have been working during the last year on a means for measuring exactly what the shock-absorber, so-called, does in the way of dissipating energy. As we have been discussing them here, all shock-absorbers seem to be regarded as very much alike.

I am not interested in any particular shock-absorber, but am able to show a difference in their action, like putting them under a microscope. Actually, all the work that the spring-control instrument can do is to transmit energy through the link between the instrument and the axle. If we could measure the magnitude of that force at every instant of the cycle, we could tell exactly the difference between two instruments or between the same instrument with different settings or with different liquids or different temperatures. I have such a method and shall be glad to explain it to anyone who wants to hear about it.

The basic principle is to attach the device to a spring that has no resistance. I made a coil spring that was sufficiently heavy and long so that I could get the equivalent of any car spring between that for a heavy car and one for a light car. That showed a difference in damping-out ability but did not show the reason for the difference; so I had to go further in that way, and I now have something which, although it is in a crude state, is just as accurate, I think, in showing what goes on in the connecting link of the shock-absorber as the steam-engine indicator is when it is attached to the steam cylinder.

CHAIRMAN GRISWOLD: — Mr. Weaver has done considerable shock-absorber testing. He worked this instrument out for the purpose of studying the comparative merits of instruments of different kinds.

O. B. WIKANDER[7]:—The riding-comfort of a vehicle is, to a very great extent, dependent upon the characteristics of its spring suspension. The best possible spring suspension is governed by such limitations and conditions as:

(1) Available travel for the spring suspension
(2) Intensity of shocks which the spring suspension is required to take up within said travel
(3) Range of static loads to which the spring suspension will be subjected in service.

If due consideration is given to these and similar limitations and conditions imposed by the service requirements, the next step would naturally be to draw a diagram of spring characteristics that would give the best possible riding-qualities. The extensive literature on the subject as well as our own investigations extending over a few years have given us what we believe to be a fairly exact picture of the ideal requirements on which the design of such a diagram should be based.

Assuming that we knew exactly the characteristics or the diagram of an ideal spring-suspension for given conditions, it is safe to say that, with the conventional types of spring—the coil spring and the leaf spring—it would be next to impossible to embody the desired characteristics in any simple combination of such springs. The coil spring, which was invented about 300 years

[5] M.S.A.E.—Eaton Axle & Spring Co., Detroit.
[6] M.S.A.E.—Automotive engineer, Trundle Engineering Co., Cleveland.
[7] Ring-spring department, Edgewater Steel Co., Pittsburgh.

ago, has, if used alone, entirely too little internal friction to serve as a vehicle suspension in most practical cases. The leaf spring, which was invented about 125 years ago, has generally too much internal friction to give the best riding-qualities on comparatively smooth roads, but never enough to give the best possible riding-qualities on very bumpy roads. This spring has, however, been adopted most generally, with resort to additional means to supply the lacking frictional resistance on bumpy roads.

With the advent of the ring spring, which is a new machine-element that has ample or even too much internal friction for any riding conditions, means are available by proper combination of this spring with a coil spring, which is comparatively frictionless, to obtain almost all desired characteristics of the spring suspension of a vehicle without resorting to any additional devices such as snubbers, shock-absorbers or stabilizers.

Conclusions Deduced from Investigations

For the last few years the manufacturers of the ring spring have undertaken extensive pioneer work to investigate whether riding-qualities heretofore impossible

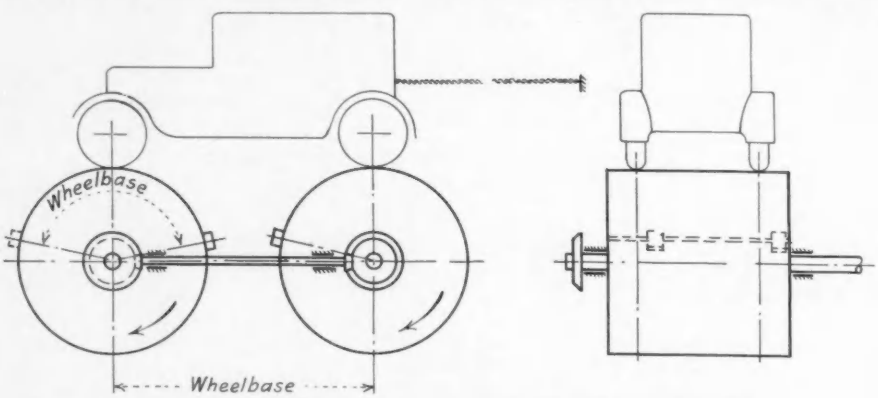


FIG. 14—SUGGESTED METHOD OF MAKING RIDING-QUALITIES TEST

The Front Wheels and Rear Wheels Would Rotate on Separate Drums Each Having a Bump. The Two Bumps Would Be Out of Phase by a Distance Equal to the Wheelbase of the Car and by Means of Suitable Mechanism Could Be Moved under the Wheels, after They Were up to Speed, for a Single Revolution, thus Simulating Actual Road Conditions

to secure could be obtained with spring suspensions consisting of suitable combinations of coil springs and ring springs. These investigations have led to the following conclusions:

(1) The small bulk of the ring spring and its cylindrical shape facilitate the enclosure of the spring in a self-contained grease-filled housing, thereby obtaining permanent and reliable lubrication of the spring and practical elimination of wear.

(2) The well-lubricated spring of this type is very uniform in its action, can be controlled exactly and the obtained riding-qualities are virtually permanent.

(3) The ring-spring suspension is considerably sturdier than the conventional suspension, tests having shown that the same amount of abuse that repeatedly breaks the conventional springs had no effect on the ring spring.

(4) This type of spring lends itself particularly well to independent springing of each wheel, which is regarded as a great step forward.

(5) Excellent snubbing or damping can be obtained with such a spring combination.

A standard make of automobile has been rebuilt and equipped with a combination of ring springs and coil springs and we have obtained excellent spring characteristics. These characteristics were obtained with a ring-spring weight of 9½ lb. and a coil-spring weight of 80 lb., giving a total spring-weight of 89½ lb. as compared with the 174 lb. of leaf-spring weight which was replaced. In comparing these figures it should be noted that the weight of the leaf springs does not include the weight of separate shock-absorbers, which are standard equipment on the leaf-spring-equipped car, nor do the weights of the coil and ring springs include the housings and mountings which they require. The riding-qualities obtained with the light coil-spring and ring-spring combination are far superior to those obtained with the standard car-construction with leaf springs and shock-absorbers.

The new form of spring has also a wide field of application to shock-absorbers. Important features are that its action is not influenced by temperature changes and that no time element is involved. The best method of applying a combination of ring spring and coil spring may differ for different car-designs, but we believe that the riding qualities of any vehicle can be improved by the use of a properly designed suspension of this type.

BORIS P. SERGAYEFF[8]:—We do not know how to exactly measure riding-quality but we do know what constitutes ideal riding; namely, that the path of the body follows a straight horizontal line. From this point of view of ideal riding-quality, I shall consider Professor Nickelsen's experiments, limiting my discussion to hydraulic two-way shock-absorbers, as suggested by the Chairman.

First, how can we compare the motion of the body registered in Professor Nickelsen's experiments with practical riding if only the rear part of the chassis was affected by the bump and the front part was permanently fixed? It is no secret that when the front wheels pass over a bump the rear part has a tendency at first to go down. When the rear wheels pass the same obstruction, conditions are different from those in the experiments.

Professor Nickelsen admits that, at a speed higher than 20 to 30 m.p.h., the oscillation of the body was not damped enough without interference to meet consecutive impulses. Such an experiment will be similar in effect to that produced by a road with carefully planted bumps at equal intervals. Such roads are not ordinarily found.

In the motion of the car going over the bump we must distinguish two kinds of vibration; free and forced. The wheel in any case must go over the bump, so the springs will be deflected and the body raised. This is the forced vibration. Ideal riding, as I defined it, will result when the springs are deflected but the body is not raised at all. This is, however, only ideal. When the wheels are on the ground but the body high in the air, free vibration starts. In Professor Nickelsen's experiments,

(*Concluded on p.* 752)

8 M.S.A.E.—Engineer mathematician, General Motors Corp. Research Laboratory, Detroit.

Rubber Articulation for Shock Absorption

By ROBERT F. COWELL[1]

CLEVELAND SECTION PAPER — *Illustrated with* DRAWINGS AND PHOTOGRAPHS

THE AUTHOR instances the automobile as being a shock-absorption machine that needs shock insulation because it travels at a higher speed each year and thereby causes greater intensity of each shock to which it is subjected, and then goes on to describe how rubber may be used in various ways to provide such insulation.

The shock insulator for use on spring shackles is a relatively large cushion of rubber almost completely housed and set up with the correct initial displacement. It is made in one piece, but has sections of rubber to cushion the spring end in all directions. The governing principle is to float the supporting members between two or more rubber cushions.

After explaining the meaning of the term "displacement" in reference to rubber insulation, the author describes the shock-insulator design for engine supports and so-called "shudder-proof" insulation for industrial machinery, afterward commenting upon a torque insulator which is a flexible-cushion connection that can be inserted in the driving line, such as between the clutch and the transmission. This is intended for use in straight-line drives only and, according to the author, it makes possible the omission of the universal-joint when so used.

Other subjects presented by the author include the use of rubber-block road-paving in which the blocks are hexagonal, 3 in. long on each side and 2 in. thick; and how rubber is used for body-panel insulation and for shock-proof seats for passengers.

THE rubber "shock-insulator" is not new, but the basic considerations which have influenced the design of this application of rubber to the motor-vehicle will make clear why this type of insulator can do its work particularly well. The car one drives is a "shock-absorption" machine. It is traveling at higher speed each year, which means that the intensity of each shock is increased. This machine must absorb not only the shocks from the road, but shocks occasioned by harsh engagement of the clutch or the sudden application of the brakes. To these must be added the vibration set up by the reciprocating and rotating parts of the different units, such as the engine or the transmission. This vibration, which is shock of lower amplitude but higher frequency, is even more destructive to the other parts of the machine. With metallic supports or connections for the units, all these destructive shocks are dissipated into the entire chassis and body, and result in systematically shaking it apart. This can be appreciated, as perhaps one of the greatest difficulties is to eliminate the squeaks and rattles.

We realize that the automobile, whether it be passenger-car, motor-truck or motorcoach, is but a self-propelled gasoline-burning contrivance for moving material quickly from one place to another. It comprises many mechanical movements and mainly utilizes metallic parts. It was our first task to make a machine which would run. Ever since then we have been trying to make that machine run more smoothly and therefore more quietly. Some have spared no expense in building complicated constructions to achieve this.

It is not peculiar that rubber is now being employed as an insulator of shock or vibration between the component units of a motor-vehicle, but it is peculiar that it was not used successfully before, since it has the ability to absorb around 50 times as much energy as steel on account of its inherent quality to withstand deformation and return to its original shape unaided. It does this work well, as the energy it gives out on expansion is less than that absorbed during contraction. Any shock or vibration transferred to the rubber causes its molecular structure to become agitated. Its low internal-friction readily allows it to adjust itself to a "constantly varying flow of shock," or to what we term vibration.

Rubber Spring-Shackle Insulation

A very popular application of rubber for shock insulation is in the shackles of the road springs of a motor-vehicle. Larger tires with lower inflation-pressure have done much to relieve the smaller shocks received from the road; however, it is very desirable to have this insulation between the sprung and the unsprung weight of a car, as many blows will be transferred through to the frame unless cushioned at these points. Any looseness such as wear of a shackle pin which allows slack in the bushing will result in sharp metallic pounding between the two.

Unlike all other applications to date, the "shock insulator" is a relatively large cushion of rubber, almost completely housed and set up with the correct initial displacement. It is made in one piece, but has sections of rubber to cushion the spring end in all directions. An individual type of cushion is used for the fixed and for the shackle ends of the spring, as shown in the upper portion of Fig. 1.

The rubber stock used was developed especially for this purpose by the rubber companies, and the compounding varies for each manufacturer. The stock as now used was adopted after making many trials before a quality was found which stood up well in service. Measured with Shore hardness instruments, it is about 68 hard and 68 elastic. It should have a minimum tensile strength of 2000 lb. per sq. in., with an ultimate

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elongation of 450 per cent of its original length. In Fig. 1 the top column over the spring in the upper view is the load-carrying column, the section below being the rebound section, and the end being the thrust section.

The area of the load-carrying column is easily computed. Tests have shown that this rubber stock will take a permanent set when the constant load carried is more than about 150 lb. per sq. in. We have selected 125 lb. per sq. in. as the basis for determining this size, knowing the normal static load.

After determining the cross-sectional area of rubber necessary to support the given load, the next step is to proportion the length to the width of the block, as illustrated in the lower portion of Fig. 1. Following our custom of providing liberal volume for cushioning effect, it would be natural to assume that the two dimensions be equal; indeed, we would do so if it were not for the fact that at this point we must inject consideration of the particular application of the cushion. We can appreciate that, to perform as a shackle at the end of a semi-elliptic road spring, it would be advisable to proportion the rubber shorter in respect to the length of the spring leaf as dimension *A* and longer with respect to its width *B*. In applying the rubber to the fixed end of the spring, in a chassis employing a Hotchkiss drive, the consideration of axle movement enters. Here the ratio of length to width of the rubber block is carried even to a greater extreme, the narrower cushion resulting in less movement; however, this ratio in either case is not made greater than is necessary for practical operation.

In the case of the fixed end we desire some movement, but it must not be excessive. If we stopped it entirely we would lose one of the best features. The reasons are that when the brakes are suddenly applied the rubber is displaced, quickly at first and with the rate decreasing as the blow is expended, allowing what would otherwise be a sharp shock to the entire mechanism of the chassis to become only an easy "give." The difference is similar to that between a blow from an iron mallet and a blow from a rubber one; the former may cause fracture and the latter will not leave even a dent. So also is the case of forward thrust occasioned by the sudden engagement of the clutch. This is dissipated in the thrust section of the rubber-cushioned block, which performs this duty very effectively.

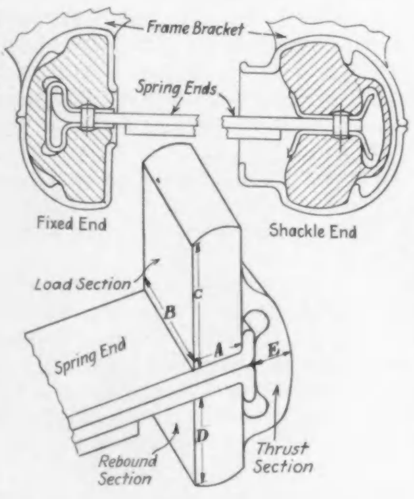

FIG. 1—RUBBER SHOCK-INSULATOR FOR SPRING SHACKLES

An Individual Type of Rubber Cushion Is Used for the Fixed and for the Shackle Ends as Shown in the Upper View. The Manner of Proportioning the Length to the Width of the Block is Illustrated in the Lower View

Support Floats between Cushions

The principle of the shock insulator has always been the floating of the support between two or more cushions. In the case of the spring shackle we have the upper load cushion above the spring leaf and a lower or rebound cushion below it.

At this point we must consider the heights of the rubber cushions *C* and *D*, in the lower part of Fig. 1. For spring-shackle work, they should be high enough to perform as a shackle. This is determined by the spring travel required. An interesting point is that the rubber cushions should be placed so that their travel will be divided equally fore-and-aft of a vertical axis, to minimize distortion. In most cases, with the spring in a flat position for normal load, this means that the cushions will be inclined toward the center of the spring one-half the normal amount of travel. Under spring deflection above or below the flat position, the rubber cushions move past the vertical position to be inclined an equal amount, this time away from the center of the spring.

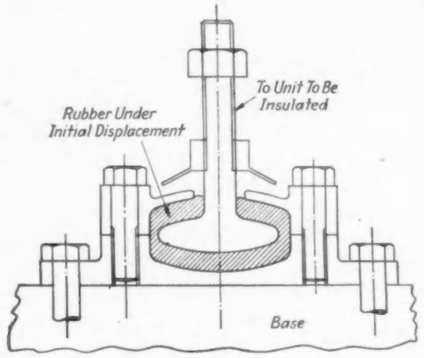

FIG. 2—SHUDDER-PROOF INSULATION FOR INDUSTRIAL MACHINERY

It Is Claimed That This Type of Rubber Cushion Has Practically Unlimited Scope in That It Can Be Used on Engine Supports, in Motor-Vehicles, Airplanes or Bodies

The cushions are made as high as it is practicable to allow for as much deflection as possible. This relieves the spring ends from extreme stresses and permits greater spring-deflection without any further increase of the motion of the sprung weight of a motor-vehicle under impact conditions. Tests have shown that the rubber stock used can be displaced until the section height is approximately one-half its original height before any permanent set occurs in the material. The height is proportioned to its length and width, in conjunction with the distortion necessary to allow the proper spring-travel. Since rubber will withstand a temporary blow of several times more than it will resist under constant pressure, the lower or rebound cushion is made less in height *D*, but no less than would be practicable for it to function as a shackle.

In addition to the two main sections, we have one which we call the thrust cushion. This, in the case of a spring-shackle rubber, is placed at the end of the spring; it takes the propulsion thrust of starting and driving. The dimension *E* in the lower view in Fig. 1 is proportioned to do this particular work and is likewise set up under the correct initial displacement to allow only a certain amount of "give." Working in conjunction with the load and rebound cushions, this thrust section performs an important duty. On front springs where complications such as shimmy, wheel fight, wabble, wandering and other evils are prone to occur, it is invaluable.

An attempt of some of the passenger-car engineers to provide a yielding fixed end by the use of coil springs at this point has perhaps been noticed. This is done to interrupt the effect of synchronism. We get our yielding automatically, without additional complications. To break up synchronism, we use the same rubber on both the right and left sides, but vary the

amount of displacement in the two by using a cavity of different dimensions. It has been demonstrated by exhaustive tests that shimmy is not aggravated by shock insulators; rather, it is counteracted by them because the rubber cushion provides a damping action.

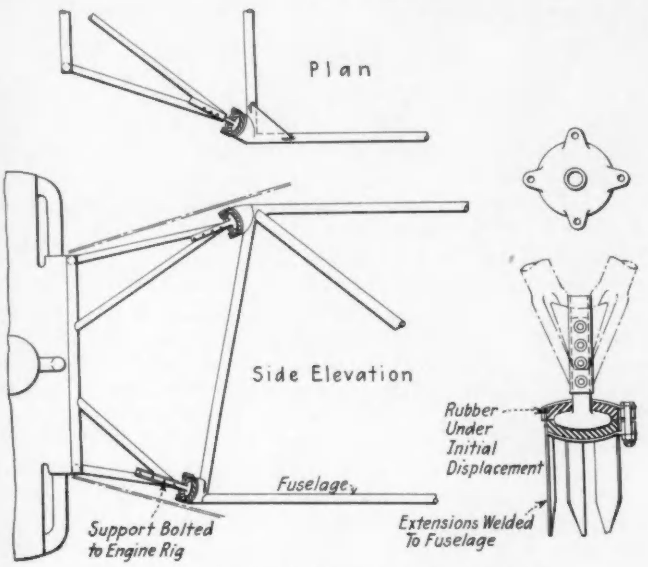

FIG. 3—AIRPLANE RUBBER-INSULATION

As Indicated, the Rubber Cushions Are Placed between the Fuselage and the Struts Which Support the Bolting Ring That Supports the Engine

Mention has been made that the cushions are set up under initial displacement, usually in a vertical direction. This is usually referred to as compression; but, as the stock is practically incompressible, we have termed it "displacement." By this we mean that the housing—in height, in particular—is made less than the combined height of the load and rebound cushions of rubber and the spring leaf and bearing plates or cups. As the rubber must flow or bulge out in the horizontal direction, sufficient space is provided in the metal housing for it to do so. This is a matter of calculating the volume.

Provision for displacement may also be accomplished by providing the same amount of space by means of holes or grooves in the rubber. The amount of initial displacement is made sufficient to assure that the spring leaf will at no time leave the surface of either cushion. Thus, the spring end must float between the two cushions, alternately displacing one and then the other. The initial displacement is made, as nearly as possible, equal to the static load to be carried so that the position of the spring leaf in the cushion will remain the same after the load is applied. The provision of initial displacement assures the constant working of the entire molecular structure of the rubber, increasing its life and at the same time keeping it very sensitive to absorption of shock and vibration.

It has perhaps been noticed that the latest types of rubber shock-insulators are of a rounded design rather pleasing in appearance. This has not been done for the sake of beauty alone, but also to secure better performance. The cushions are rounded on the upper and lower ends and the sockets of the housings, into which the rubber fits, are also rounded but have a larger radius. This results in greater pressure at the center of the column than at the edges, allowing it to shackle more easily.

It should be said that the shock insulator is a shock cushion which irons out the irregularities of the road and saves the chassis and body a great deal of punishment. A sufficient volume of rubber must be present to accomplish this satisfactorily. When shock insulators are used, the passenger is conscious of an easier and a quieter ride.

Shock Insulators on Engine Supports

Reciprocating and rotating parts within the engine set up vibration which is very destructive to the chassis and body. In applying the shock insulator for this use several problems not found in the spring shackle were presented, because in this case the connection had to be interposed between rigid arms and stiff side-frame members, thus permitting the engine to float, but not to float too much. Too great movement of the engine was overcome by providing ribs on the supports which enter grooves in the rubber cushions. This divided the rubber into a number of small sections through which no great amount of movement could be obtained. In this way enough volume of rubber could be used to give the proper absorption. At this point, tests convinced us that, while a little too much rubber was not

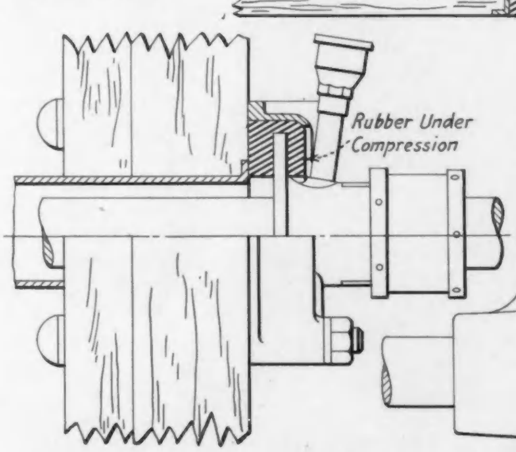

FIG. 4—INSULATION OF A MARINE-TYPE STUFFING-BOX

Any Vibration Carried by the Shaft Is Prevented from Being Transferred to the Bulkhead and to the Other Parts of the Hull

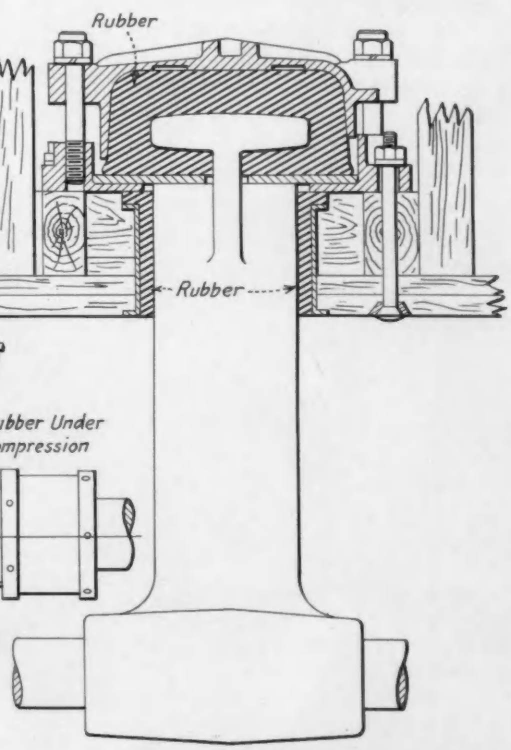

FIG. 5—RUBBER-INSULATED STRUTS FOR SUPPORTING THE PROPELLER-SHAFT BEARINGS

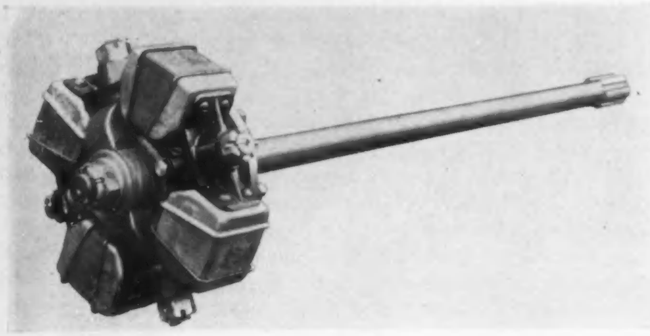

FIG. 6—RUBBER TORQUE-INSULATOR FOR USE IN A DRIVING LINE

It Consists of Four Generously Proportioned Cushions of Rubber Assembled into Cups Mounted on Two Arms, the One Driving and the Other Driven. The Arms Float between the Rubber Cushions

detrimental with this new design, less rubber could be used to damp effectively this high-frequency vibration of low amplitude.

Shudder-Proof Insulation

The results of numerous experiments culminated in a standardized line of what we call shudder-proof insulation for industrial machinery, an example being shown in Fig. 2. A wide range of standard rubber-insulators and companion parts have been developed to meet the needs of practically any application. This type of rubber cushion has practically unlimited scope. It can be used on engine supports, in motor-vehicles, airplanes or boats. Its application to stationary machinery is saving wear and tear on the equipment, the building and the employes. It can be used wherever the transfer of vibration from one source to another must be stopped, and the insulator is not ordinarily required to take care of any movement in such instances.

Briefly described, this new rubber cushion is made in two parts for convenience of assembly. It could be made in one piece if the support could be forced through the small hole, but we found that it was better to make it in two pieces. It cushions the support in all directions. The metal container is made in two pieces as formerly, and initial displacement is provided. The end of the support which enters the rubber is curved, as is also the rubber and the socket in the housing allowing for placement at different angles. Grooves are molded into the rubber to take care of the initial displacement and to allow the structure of the rubber to flow during the process of absorbing the vibration. It may be said also that the rubber on all corners is rounded, a larger radius being used than for the radii in the sockets or the container. This leaves room for the rubber to flow. It is shaped so as to distribute, as evenly as possible to the entire cushion, the maximum internal stresses set up by the shock.

Use on Airplanes and Boats

For airplane-engine supports, the rubber cushions are placed between the fuselage and the struts which support the bolting ring that holds the engine, as indicated in Fig. 3. Four planes have been equipped with this construction, and the results have been very satisfactory; the minimizing of vibration and noise is very beneficial in this application.

It is necessary to apply the cushions between the engine and the engine-bed stringer, when using rubber insulation on a boat, and the shudder-proof-insulator construction is employed for this purpose. Besides the engine, the entire line of drive through the hull must be insulated to achieve complete results, and an insulated stuffing-box was therefore developed as indicated in Fig. 4. Any vibration carried by the shaft is prevented from being transferred to the bulkhead and to the other parts of the hull. The struts for supporting the remaining propeller-shaft bearings are also insulated, as shown in Fig. 5. Rubber shock-insulators have proved themselves very effective on power yachts; they reduce vibration and noise to the minimum and add greatly to the comfort of the persons on board.

Prevention of Chassis Distortion

In a motor-vehicle which is shock insulated at the springs and has shudder-proof insulation at the engine and the transmission supports, if the latter be amidships, the other units of the chassis do not need to be protected so much from vibration as they do from frame distortion. For this application we use our first engine-support design, which had larger cushions of rubber. They prevent the weaving of the frame from throwing great stresses into these units and, if necessary, will deaden any vibration. We find use for this type for the radiator, the cab, the gasoline tank, between the outriggers of the frame and the body of a motorcoach, and for the torque arm, the distance rod and the like; but in some instances special conditions had to be met by special means.

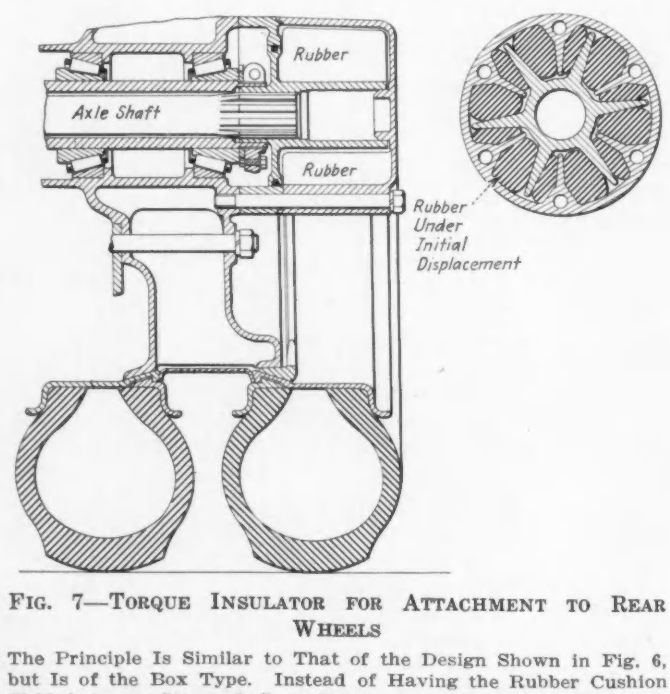

FIG. 7—TORQUE INSULATOR FOR ATTACHMENT TO REAR WHEELS

The Principle Is Similar to That of the Design Shown in Fig. 6, but Is of the Box Type. Instead of Having the Rubber Cushion Held between Stamped Cups Fastened to Two Arms They are Retained between the Ribs of Two Cast Housings, One of which Is Bolted to the Axle Shaft and the Other to the Wheel Hub

Description of Torque Insulator

The torque insulator is a flexible-cushion connection which can be inserted in the driving line, such as between

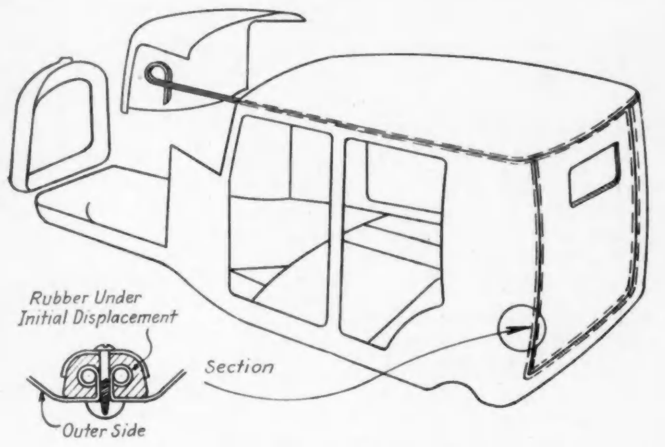

FIG. 8—RUBBER BODY-PANEL INSULATION

The Idea Is To Prevent Vibration and Drumming of Closed Bodies. The Fact That the Bodies Have Edges which Are Turned Permits a Rubber Molding To Be Snapped into This Shaped Portion and Put under Displacement by Clamping. Body Panels Are Thus Held Together Semi-Rigidly, and Will Not Squeak, Vibrate, Rattle or Drum

the clutch or transmission, and is shown in Fig. 6. It consists of four generously proportioned cushions of rubber assembled into cups which are mounted on two arms, the one driving and the other driven. These rubbers are assembled with initial displacement; the arms float between the cushions, which are never wholly free from initial displacement as in the case of the spring-shackle-rubber design. One trouble experienced at first was that of the effect of centrifugal force. To overcome this, the volume of each rubber was reduced and the shape of the retaining cups was changed to neutralize the effect of this force and obtain a torque moment that was tangent to the radius of action.

The torque-insulator assembly is intended for use in straight-line drives only, although it is able to take care of a reasonable amount of misalignment or universal-joint action. This makes it possible to omit the universal-joint where the insulator is used. Sudden excessive shock or torque, such as is occasioned by dropping the clutch in, is absorbed in the rubber cushions because of the temporary distortion of the rubber. This means that the "flow of torque" is brought from small to large proportions, or vice versa, in a smooth manner and without damaging the transmission gears or the shafts.

Another function of the torque insulator is the prevention of the passage of vibration from one unit to another, such as from the engine to the transmission. This checks the possibility that the two units will vibrate in unison and produce resultant forces of serious amplitude. At the same time, high-frequency vibration of low amplitude is constrained within the cushions. All this prolongs the life of the chassis and of the body of the vehicle and gives greater comfort to the passengers.

We have recently developed a torque insulator to be attached to the rear wheels, as shown in Fig. 7. The principle is similar to that of the design used in the driving line, but is of the "box-type." Instead of having the rubber cushions held between stamped cups fastened to two arms, they are retained between the ribs of two cast housings, one of which is bolted to the axle shaft and the other to the wheel hub. The rubbers are assembled under initial displacement, and this is accomplished by tapering the cushions and wedging them into place.

At first thought, it seems that this application is hardly warranted, because the tire is capable of accomplishing this effect to some extent and especially so if pneumatic tires are used. But for dump trucks working in excavations, where the drivers are constantly dropping the clutch in quickly to rock the truck out of a rut, it is a great saving to axle shafts and gears, to the transmission and to the engine. For instance, in a truck having a final ratio of 7.6 in low gear, a ratio of 5.3 gives a total reduction of over 40. A 9-deg. "give" in the torque-insulator cushions is equivalent to one revolution of the engine. We do not actually get a 9-deg. "give"; it is about 5 or 6 deg., but that is considerable. It should be noted that the cushions are of different sizes. The cushions for forward drive are much larger than those for use in reverse drive. That was done to get the maximum motion possible, the idea being that the reverse drive is not as important.

Use of Rubber for Pavements

Another application of the shock-insulator principle is one which, while not used directly in a motor-vehicle, may influence its design. This is rubber road-paving. The rubber cushions are of hexagonal shape, being 2 in. thick and having a length of 3 in. on each side. Each rubber is held under initial displacement by forcing it into place in a hexagonal band 1¼ in. wide by 3/16 in. thick, formed from 6-in. tubing and galvanized. They are laid the same as granite block, using a grout of sand and tar, and preferably on a concrete base 6 in. thick.

These paving blocks can be made from reclaimed rubber,

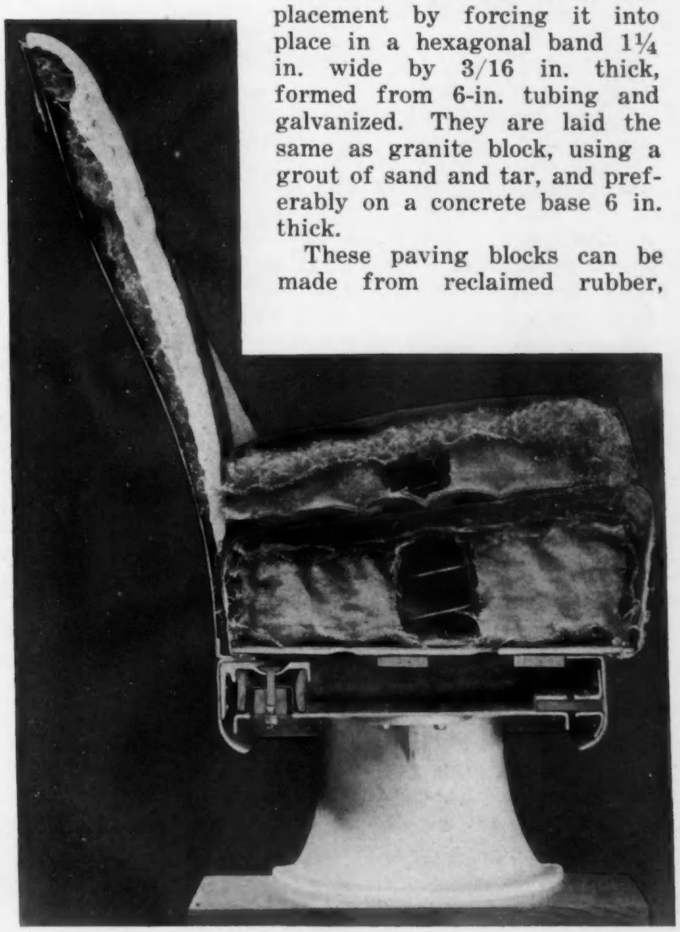

FIG. 9—SPECIALLY DESIGNED SEAT-FRAME, SHOCK INSULATED WITH RUBBER UNDER INITIAL DISPLACEMENT

The Seat Consists of the Seat Frame and Back, the Rubber Shock-Insulators and the Supporting Plate or Pedestal

and they open a larger field for this material. The cost of laying such a road is approximately three times that of granite block, but this is offset by the fact that it will last three times as long, and has the advantage of being quiet and less damaging to vehicles.

Rubber Used for Body-Panel Insulation

Rubber shock-insulation is especially adaptable for insulating closed bodies to prevent vibration and drumming, the method being shown in Fig. 8. The fundamental idea is to utilize the fact that the bodies have edges which are turned, snap a rubber molding into this shaped portion and put it under displacement by clamping. Body panels are thus held together semi-rigidly; they will not squeak, vibrate, rattle or drum.

Present-day large-quantity production-methods are speeded up by the use of rubber shock-insulation in bodies. Panels and stampings do not need to be so accurate as when welded or riveted construction is used. Bodies formed of separate panels can be held together tightly and yet be semi-flexible. Rubber shock-insulators are equally effective on bodies made with individual stampings or the all-metal stamped bodies with each side made from a single sheet. They give metal bodies the advantages of the Weymann body, without sacrificing rapid production.

Rubber-Insulated Seats for Passengers

Until recently, spiral springs under the upholstery of passenger seats have been the only means in use for the absorption of vibration at that point. The realization that a gentle motion of the seat conduces greatly to passenger comfort has led to a specially designed seat frame, shock insulated with rubber under initial displacement, as shown in Fig. 9. This seat consists of three units: the seat frame and back, the rubber shock-insulators, and the supporting plate or pedestal. A number of specially designed rubbers are fastened to the seat frame and rest on a supporting metal structure. This arrangement absorbs shocks and vibration, at the same time allowing for maximum flexibility of the seats and permitting them to recede slightly under weight so as to provide additional leg room. This construction has been demonstrated in service to afford exceptional comfort; it absorbs both road shocks and engine vibrations. Rubber shock-insulated seats respond to the movements of drivers and passengers, and contribute a rhythmic action which is conducive to comfortable motoring.

These applications of rubber in the automotive field cover a wide range, and we are confident that it will not be long before a better appreciation of their value will cause their more general adoption.

Shock-Absorbers

(Concluded from p. 746)

shock-absorbers are forced at the rate of 100 cycles per min. In the car, free vibration will be at 75 to 125 cycles per min. To think that the force of the hydraulic shock-absorbers will be exactly proportional only to the square of the linear speed of the fluid is, I suspect, a little too theoretical; and we know that theory very rarely coincides with practice.

In the forced part, impulses will be of the order of 5,000 per min., so resistance will be of the order to break the linkage or shear the bolts or to break out a piece of the frame, as I have seen actually happen. This is not at all exaggerated.

Suppose the car is passing at a speed of V m.p.h. over an obstruction $2L$ in. long. The time required to climb such a bump will be

$$0.05682 \times L/V \text{ sec.}$$

which is the equivalent to the experimental speed

$$528 \times V/L \text{ r.p.m.}$$

For example, at a speed of 30 m.p.h. with an obstruction 6 in. long, the time will be 0.005682 sec.; that is, 5280 r.p.m. At such a speed we may expect only a very heavy preloading, and the chassis will consequently deviate very far from the ideal path.

Professor Nickelsen's suggestion of adjusting the front and rear damping may be justified in practice, but I do not see any reason for that in his experiments. I would, however, suggest that his experiments be modified to obtain more exact information on riding-qualities and on the adjusting of the two-way hydraulic shock-absorbers.

Fig. 14 shows a new arrangement. The car is placed on two drums connected by a 1:1 gear. On each drum is fixed a bump, hole or what not in such a way that the two bumps are out of phase by a distance equal to the wheelbase of the car, so that the front and rear wheels pass over the two bumps consecutively, as they do on the road. These obstructions should be made removable by some sort of trigger so that, after developing the required experimental speed on the even surface of the drums, obstructions will be introduced for only one impulse and then removed. Thus the car will have free vibration without any interference. It is understood that the car will be held in place by a long rope.

It seems to me that an experiment thus conducted will simulate exactly the behavior of a car on the road and yield extensive information from the laboratory test.

Developing a Front-Drive Car

By W. J. MULLER[1]

SECTION MEETING AND ANNUAL MEETING PAPER

Short papers on the subject of front-drive cars were presented by Mr. Muller at meetings of the Metropolitan and Cleveland Sections and at the Annual Meeting. These papers have been combined herein and the discussion at the three meetings follows. Emphasis is given in the paper to the importance of the universal-joint in front-drive designs and to the suitability of the joint adopted on the car designed by the author. Mention is made of the transmission used, in which the worm drive is located centrally to reduce the distance between the axle and the clutch. The lack of an engine designed particularly for the needs of a front-drive car is mentioned. Questions of weight distribution under static, hill-climbing and acceleration conditions are discussed, and driving experiences are recounted.

The discussion was participated in by leading engineers of the industry. That at the Metropolitan Section meeting had to do with several fundamental theoretical considerations and practical problems, including lubrication and the minimum desirable height of cars. The Cleveland discussion centered largely in driving and designing experiences of engineers not directly interested in front-drive cars.

Engineers having designing experience with other front-drive cars participated in the discussion at the Annual Meeting, as well as other prominent engineers in the industry. Questions of front-wheel setting and tire wear and of the relative desirability of front and rear-drive cars from various points of view were given much attention, and a suggestion was made for worm-drive lubrication.

GREAT interest has been aroused in this Country among automotive engineers, manufacturers and the general public by the appearance of front-drive passenger-cars. Natural questions to ask are: What are the reasons for the front-wheel-drive automobile; and what are its advantages and disadvantages as compared with those of rear-drive cars?

No serious attempt had been made to develop front-drive passenger-cars in this Country until a few years ago, when the producers of the two cars of the type now on the market began work upon them. As to the relative merits of front drive, opinions have differed, because insufficient data were available to make a fair comparison, such as might be based on extended experiments on the two types under identical conditions and extending over a period long enough to allow time for necessary improvements in the newer design.

This paper is written to share with other engineers the impressions and experiences gained in developing the Ruxton front-wheel-drive car.

The Ruxton car really was built up around the Weiss universal-joint, which has been described previously.[2] This joint offers two features that were considered essential for the purpose: It transmits absolutely uniform motion, even at extreme angles, and it functions efficiently under conditions of indifferent lubrication. The ordinary univeral-joint, when transmitting motion between shafts making an angle of 30 deg. from a straight line, gives to the driven shaft a total fluctuation of speed of approximately 30 per cent. This fluctuation can be neutralized in a front-wheel drive by the use of two joints at the wheel, with an arrangement for dividing the angular deflection equally between them.

The use of rolling-ball contacts for the articulated members of the Weiss joint is a vital point. While lubrication is still necessary, as for ball-bearings, the necessity for it is reduced to the minimum, and all danger of seizing is eliminated. It sounds well to say that a universal-joint has an efficiency of 99 per cent, but a loss of 1 per cent of 100 hp. means that 42 B.t.u. per min. is being transformed into heat. This is faster than small parts ordinarily can conduct or radiate heat, and is equivalent to raising the temperature of 10 lb. of iron 42 deg. fahr. for each minute of operation. At that rate, with no radiation, the flash point of lubricating oil would be reached in less than 10 min.

The great obstacle to front-drive development is the lack of a suitable engine, built definitely for front-drive cars, having the fan drive and other accessories arranged suitably for the reversed position of the engine. It is our opinion that, when a powerplant is properly disposed, front drives require no longer wheelbase than rear drives having the same amount of body room, and that any engine manufacturer who will develop such a unit will find a ready market for it and will help to stimulate interest in front-drive cars.

Pulling Adds to Car Stability

The advantages of driving by the front wheels have long been recognized. It is significant that the first airplanes were pushers, and their instability was one reason for reversing airplane engines so that nearly all our airplanes today are tractors. Drivers of racing automobiles also have found greater stability in pulling the cars from the front wheels than in pushing them from the rear. This is particularly noticeable on turns, when the driving effort is applied in the direction of the turn, with a front-drive car, and at an angle to the position of the front wheels, in a rear-drive car.

[1] M.S.A.E.—Vice-president, chief engineer, New Era Motors, Inc., Philadelphia.

[2] See THE JOURNAL, July, 1924, p. 96; and December, 1926, p. 632.

Effects of this sort are more noticeable at the higher speeds of airplanes and racing cars; however, we have noticed a greater smoothness of turning, especially in hill climbing and in parking, with the front-drive design.

We anticipated some difficulty in negotiating certain steep mountains in driving from Philadelphia to Indianapolis and back by way of Pittsburgh. I have driven over this road many times with rear-drive cars. The front-drive car seemed to be an equally good hill-climber, and it seemed to pick up speed on the turns where the rear-drive cars would slow down. I cannot say definitely what is the cause of this, but I believe that the smaller spring deflection at the driving axle has much to do with it. I believe that this small deflection also influences the unexpectedly small wear on the front tires, which indicates absence of tire slippage, and it seems to make unnecessary so large a differential as is needed in a rear-drive car.

Perhaps the greatest departure from conventional details in the Ruxton car is in the transmission. Obviously, the center of the differential should be on a line between the front wheels. Placing a conventional transmission and a straight eight-cylinder engine between this point and the dash results in too long a car. Our solution has been to insert the worm drive in the middle of the transmission, thus making a saving in length of about 6 in. Gearshifting is effected by means of a horizontal rod with a handle placed just below the instrument board. The arrangement is such that the shifting movements correspond to those of a gearshift lever using S.A.E. Standard positions. The front axle, which is of I-beam section, is bowed forward between the springs to clear the transmission, and the radiator also is built around the transmission.

We have found the total weight and the weight distribution to be favorable with the front-drive car. Our five-passenger sedan, having a wheelbase of 130 in., weighs only 4000 lb. when completely equipped and filled for the road but without passenger load. The weight on the front wheels is 2125 lb., and that on the rear is 1875 lb. The unsprung weight is 390 lb. at the front and 280 lb. at the rear. The total weight is less than would be required for a corresponding rear-drive car without sacrificing rigidity.

Steering Action and Traction Are Good

Neither shimmy nor wheel fight have been experienced in developing the Ruxton car; the drive does not affect the steering at all. The driving elements have received more study than the rest of the car; because we realized that a poorly designed drive might cause much trouble with the steering, and it was necessary to dispose a large amount of machinery in a small space.

Our first thought was that the center of the outer universal-joint must be absolutely in line with the steering-knuckle pivot, and we went to some trouble to place it there. We found that the early models were so perfectly balanced that the steering was dead and the cars had a tendency to wander. The remedy was found by simply moving the center of the universal-joint far enough from the axis of the pivot to create a slight drag. It is possible to have this distance quite considerable before the driving torque causes any annoyance.

Various people have stated that one disadvantage of the front drive is a loss of traction on steep grades. It is true that there is such a loss, but the amount is so slight as to make the question merely academic. On a 15-per cent. grade, the center of gravity of the Ruxton car is moved back about 4 in., which is about 3 per cent of the wheelbase. A traction loss of 3 per cent seems to be of little importance when we consider that the grip of the tires on the road under different conditions varies between 25 and 100 per cent of the load.

Another point of academic interest is the transfer of weight during acceleration. Few cars have actual acceleration greater than 3 ft. per sec. per sec. on high gear. With the same center of gravity and wheelbase, the net loss of weight on the front axle amounts to 69 lb., or 3½ per cent of the static load, for acceleration of that order. The question of load transfer during deceleration, when four-wheel brakes first occupied our attention, was a real issue; it is of minor importance in connection with front-wheel drives.

Front-wheel driving has many advantages, and I believe we shall see a trend toward this construction when its advantages are more widely recognized. There is nothing inherently costly in the design; on the contrary, economies appear which even now make it apparent that a front-drive car can be built for less than could an equivalent conventional rear-drive car.

DISCUSSION AT THE METROPOLITAN SECTION MEETING

P. M. HELDT[3]:—In discussing traction and skidding of cars, we must distinguish between front-wheel skids and rear-wheel skids. We are more familiar with rear-wheel skids.

What ordinarily holds cars in control while turning corners is the difference between the resistance to rolling motion and the resistance to sliding motion. On a normal, good road, the resistance to rolling motion of a wheel is about 20 lb. per 1000 lb. of load and the resistance to sliding motion about 600 lb. per 1000 lb. Anything that decreases this difference increases the possibility of skidding.

Locking the rear wheels, by means of brakes, has the effect of making the resistance to motion equal in all directions, and a skid is very easily induced in this way.

The rear unsprung weight is much less in a front-drive than in an ordinary rear-drive car; that tends to make the rear wheels hold the road better, thus improving the directing tendency of the wheels.

HERBERT CHASE[4]:—Because some front-wheel-drive layouts require long wheelbases, it does not necessarily follow that all front drives will do so. One alternative is to place the transmission in front of the axle, and other alternatives may be developed.

Racing drivers and others experienced with front-wheel drives report that, while it is possible to negotiate curves at higher speed with front than with rear drives, serious skids are likely to result if the throttle is closed while the car is in the turn.

Maximum tractive ability depends chiefly upon weight distribution. If all other factors are equal, the car with the greatest proportion of weight on the driving wheels,

[3] M.S.A.E.—Engineering editor, *Automotive Industries*, Chilton Class Journal Co., Philadelphia.

[4] M.S.A.E.—Associate editor, *American Machinist* and *Product Engineering*, McGraw-Hill Publishing Co., New York City.

whether front or rear, will have the greatest tractive ability.

Wear of tires on the front wheels of front-drive cars has been excessive in some cases, but the fact that the wear is not excessive in other cases indicates that this is a result of some other factor than the front drive itself.

Front-drive cars having brakes adjacent to the differential housing have a distinct advantage, over cars otherwise similar but with brakes on the wheels, in respect to low unsprung weight. In my opinion, however, the maximum advantage of the front drive is obtained only when the wheels are individually sprung and all brake mechanism is removed from the wheels.

W. J. MULLER:—I think the location of the drive has nothing to do with the question of skidding; that depends on weight distribution and balance. We have driven fast on turns and shut off the engine without skidding.

Voicing a Call for Gasoline Economy

F. H. DUTCHER[5]:—The performance of a car should be measured by something other than hill-climbing stunts. Transportation means getting somewhere with reasonable comfort, at a reasonable speed and cost.


P. M. HELDT

Recently I placed a vacuum gage on the inlet manifold of an ordinary car. It indicated a depression in pressure of 14 in. of mercury at speeds between 20 and 40 m.p.h. on a level road in fairly still air. That corresponds to about one-half the friction of the engine and to about one-quarter or one-third of the load on the engine. That is why we usually get only about 25 ton-miles per gal. of gasoline with our cars, instead of 50 or 60 ton-miles per gal.

Gasoline is cheap now, but it will cost more some day. I should like to see more work being done on cutting down the fuel cost of operation. Improvement in the comfort of passengers is one of the tendencies of the day.

GEORGE W. DUNHAM[6]:—The front-wheel drive offers a considerable opportunity for reduction in unsprung weight, which is of great importance for riding comfort. Reducing the weight of the rear axle is a big step in that direction. Lowering the center of gravity reduces the uncomfortable swaying sensation observed in so many cars at speeds of 40 or 50 m.p.h. While I think that the average person does not drive at high speeds, he desires to be able to ride at such speeds in comfort on occasions when he has a large distance to cover.

There is no doubt that applying the power at the front end, where much of the mass is concentrated, tends to pull the car around a curve without skidding.

Front Driving Alters Car Action

W. S. JAMES[7]:—Mr. Heldt offered a very reasonable explanation of the action of the rear wheels in relation to skidding. When turning a corner, centrifugal force acts along the general direction of the axle, reducing the margin between the 20 and the 600 lb. per 1000 lb. which gives directional control. The maximum driving effort is about 100 lb. per 1000 lb. with maximum acceleration on high gear, about 200 lb. in second and about 300 lb. in first. Thus the difference in resistance between the direction of rotation and any direction of transverse slipping becomes less and less, until the wheel has no sense of direction when the forces in the two directions balance. The action of the driving wheels in this respect is the same whether they are the front wheels or the rear wheels.

I was told to keep the throttle open and maintain my speed on the turns when I drove a front-drive car recently. I practiced four or five times on one turn on a gravel road until I felt positive I could hold the speed. I do not know what my speed was, but it was a little over 65 m.p.h. when entering the turn. Before I was half-way through the turn something happened; I do not know whether I did just the right thing, but I came out of the turn sidewise. My experience was that the car did not behave just like the cars to which I was accustomed, and the experience was somewhat disconcerting.

When four-wheel brakes first appeared, drivers were inclined to throw their passengers through the windshields; now they are trying to throw them through the windows. I think these cars should have hand holds on the roof and on other favorable positions, so that the passengers can hold themselves in.

Front-drive cars have been built lower than ordinary rear-drive cars. This also is helpful in making turns at high speed, and I think the effects of the front driving and the low center of gravity have not been segregated; possibly a rear-drive car with an equally low center of gravity might make turns just as well.

Low unsprung weight is of relatively less importance as road surfaces get better; unsprung weight is most objectionable in traveling at high speed over a rough road. If extremely low unsprung weight is desirable, it is possible to go still further than the design under discussion, by omitting the axles entirely and having independently-sprung wheels, thus reducing the unsprung weight to the minimum.

Mr. Muller has outlined some interesting points in his paper, and he has a very practical car. The final answer will be made by the riding public when it decides whether or not to accept a front-drive car, with the advantages and disadvantages which it has, in preference to cars of the present type, with possible improvements that may have been suggested by the front-drive designs.

QUESTION:—What is the turning radius of the Ruxton car?

MR. MULLER:—The maximum turning angle on our front wheels now is 32 deg. This makes it possible to turn in a circle between 38 and 40 ft., which is much less than the average turning circle of a car having a wheelbase of 130 in. Soon we expect to have universal-joints allowing a little more deflection, to allow the front wheels to turn to 34 deg.

FREDRIK LJUNGSTRÖM[8]:—Mr. Muller spoke of the bet-

[5] M.S.A.E.—Instructor, department of mechanical engineering, Columbia University, New York City.

[6] M.S.A.E.—Consulting engineer, New York City.

[7] M.S.A.E.—Research engineer, Studebaker Corp., South Bend, Ind.

[8] Direktör, Aktiebolaget Spontan, Stockholm, Sweden.

ter pick-up of front-drive cars on curves in hill climbing. The explanation of this seems simple to me. It is because the front wheels travel a considerably greater distance than the rear wheels on a sharp curve. If the front wheels were turned to be at right angles to the rear wheels, the rear wheels could not drive the car at all, while the front-wheel-drive car would make the turn easily.

Securing Economy by Weight Reduction

ALFRED M. NEY[9]:—Mr. Dutcher has raised the vital question of the over-all efficiency of an automobile. Few of us stop to think that we are accelerating, decelerating and pulling up grades a mass of 4000 to 6000 lb. to transport a useful load of only 600 lb., which is the average weight of four passengers. If the car could be made much lighter, we could obtain the same performance with a smaller engine and the whole would be more efficient. I believe that front-wheel driving will make a contribution in this direction. Riding comfort depends upon the proportion of sprung to unsprung weight. If the axles are heavy, the sprung weight must be high to give comfortable riding. Independent springing of the wheels will reduce considerably the unsprung weight and will give excellent riding quality to a car having the sprung weight also reduced in proportion.

The frame and the entire rear part of such a front-drive car would not need to be built so heavily as they are for the conventional type, as they are not subject to the same stresses and twisting.

Installing the entire powerplant in front and lightening as much as possible the rest of the chassis and body will shift the center of gravity forward, as is essential with front driving to obtain maximum traction. Omitting the dead front and rear axles probably makes possible the least expensive construction. Besides, there is no possibility of shimmy in a car having independently-sprung front wheels.

We know that applying the brakes during a turn may cause a rear-drive car to skid and roll. The same thing can happen with a front-drive car, and the braking effect of the engine is applied to the front wheels if we take our foot off the accelerator. However, it is not necessary to leave the engine wide open and make the turn at 60 m.p.h. to prevent skidding with a front-wheel-drive car. If we wish to slow up on a curve, the only precaution necessary is to throw out the clutch at the same time we apply the brakes.

CHAIRMAN GEORGE A. ROUND[10]:—Ground clearance interests me less than spine clearance. I find that most modern cars have the roofs so low that I file down my backbone every time I get in. I should like to see more clearance in this direction.

D. G. ROOS[11]:—It is surprising to me that, in view of the great activities abroad on independent wheel suspension, little progress has been made in this Country since the War, and that independent suspension has found no advocates. We believe that much can be done to develop chassis design, and that much attention can be profitably given to new chassis designs and spring suspension. The specifications of all American cars are too nearly alike.

I see no reason for becoming greatly excited over the fact that some cars are being built with front-wheel drives. We have seen front-drive cars, in the races at Indianapolis, competing for several years with rear-drive cars having the same piston displacement and similar powerplants. None of them has crossed the tape a winner, in spite of what is said about the ability of front-drive cars to take turns. I do not say that this is an argument against front-wheel driving; rather, it indicates that many other items than the means and location of the drive enter into the make-up of an automobile.

Development of Rear-Drive Cars Stimulated

We have been experimenting with independently suspended wheels. We find it possible to make a car with lower unsprung weight than that of any existing front-drive car and to make the chassis just as low or a little lower. It is a question how low a car should be. Chairman Round has just raised that point, and many of us can remember several recent instances in which manufacturers have been obliged to increase the height of their cars after making them too low. A car that stands lower than 68 or 70 in. from the ground to the roof is liable to criticism because the top rails of the doors are too low. I think that a decided reaction will be felt by manufacturers who try to build cars materially lower than that.

Another thing that we investigated is the question of relative weight distribution during acceleration and deceleration, with front-drive and rear-drive cars. We have compared one of the leading front-drive cars with a rear-drive car that is identical in weight and piston displacement, and we find 25 per cent less available tractive effort in the front-drive car, because of its weight distribution. Under ordinary conditions this is not noticeable, but it is very noticeable in comparative tests for maximum acceleration in second speed or on gravel, clay or wet pavement.

D. G. ROOS

Not everything possible has been done yet with chassis of the conventional type. Front-wheel drives will produce one effect that I welcome: it will stimulate manufacturers of cars of the conventional type to develop all the possibilities of their chassis.

MAURICE WALTER[12]:—No one who has witnessed a recent Indianapolis race can help admiring the smooth way in which the front-drive cars make the turns, as contrasted to the hopping action of the rear-drive cars. No front-drive car has won in one of those races because they go around the turns wide open and the engine gets no rest. Failures of the front-drive cars have been reported as engine failures in every case. Rear-drive cars are driven with the engines throttled on the corners, thus giving them a rest which the engines of front-drive cars cannot have.

[9] Jun.S.A.E.—Designer of racing cars, C. H. Matthiesen, Jr., New York City.

[10] M.S.A.E.—Assistant chief, engineering division, Vacuum Oil Co., New York City.

[11] M.S.A.E.—Chief engineer, Studebaker Corp., South Bend, Ind.

[12] M.S.A.E.—Chief engineer, Walter Motor Truck Co., Long Island City, N. Y.

Without doubt, the front-drive car has traction limitations because of the lessened weight on the front wheels during hill climbing and acceleration, but that can be overcome by providing differentials that do not equalize the torque but divide it according to the traction of the driven wheels.

Potential Advantages of Front Driving

A. C. WOODBURY[13]:—For years I have been interested in the idea of front-drive cars because of the possibility of keeping virtually all of the mechanical parts of the car in a single, compact unit. Such a unit could be built complete and shipped anywhere, the rear axle, the chassis frame, if any, and the body to be made according to the conditions in the district or country where the car is assembled and used. Undoubtedly the industry has advanced beyond the point where this idea would have any great value unless it might be for shipment to remote countries.

Front driving seems to have a number of important incidental advantages which naturally are only partly realized in the first designs that have been placed on the market in this Country. The first group has to do with riding qualities. The considerable reduction in unsprung weight is important. There seems to be a great possibility, too, of increasing the deflection of the rear springs instead of using all of the space gained in reducing the height of the car. This will increase the vibration period at the rear and probably will make possible a slower vibration period at the front also without danger of synchronism between the front and rear springs when the rear seat is vacant. Independent springing of the front wheels and mounting the front brakes on the transmission have advantages that can be easily realized with front-wheel drives.

A quiet next-to-top gear is one of the sensible demands of the day, which is being met in a rather complicated and imperfect way by internal-gear transmissions. The double-direct drive at the rear axle was offered some years ago to meet this need, and its failure to live was due largely to its location. Built into the transmission, as is the final drive of a front-drive car, the double-direct drive should meet the present need in an ideal way. The units necessary to use with it to produce an ideal four-speed transmission are equivalent to the gears and shafts of a transmission giving two speeds forward and one reverse. This part of the transmission can be made transverse, if desired, thus making it possible to locate the cylinder block in approximately the same position as in a rear-drive car.

THE DISCUSSION AT CLEVELAND

W. E. ENGLAND[14]:—The front-wheel-driven passenger-car has an outstanding advantage in that it is possible for the body to be lowered to a point beyond which the passengers cannot see over the hood. Never before was the body designer given such a free hand, but it would be foolish to think that we will drive our future cars through periscopes. However, the type has two serious disadvantages in the lack of frontal area for the radiator and the increased length of hood and wheelbase, with the attendant increase in cost unless certain things are done.

W. E. ENGLAND

Our company, like every other, is interested in any new development, and we have investigated the possibilities of this drive as it would apply to our product. As you know, this is a high-powered straight-eight luxury car, in which the passengers are not crowded but have plenty of leg and head room, and three large people can sit comfortably in the rear seat. It is vital that such a car should ride exceptionally well, which requires plenty of spring clearance, so that long, soft springs can be used. To secure the low vertical height that this demands, a worm-drive rear-axle is used, which allows a 71-in, over-all height for a five or seven-passenger car ready for the road but with no passengers.

The car has ground-clearance of 7¾ in. under the worm-drive axle, 9¼ in. under the front axle, and 9¼ in. under the flywheel housing. The engine is set on a 4-deg. angle and completely fills the hood. It is possible to lower the whole car 1½ in. by taking out some of the cushion springs at the center of the rear seat, but the passenger sitting in that position would not ride as comfortably as the ones at the ends. The front seat cannot be lowered more than 1½ in. without seriously impairing the driver's vision over the hood.

A Study of an Existing Model

Now what happens to a car of this type when it is made into a front-wheel drive? A worm drive very similar to that of the Ruxton was laid out, having second and third speeds at the rear of the worm and first and reverse in front. Every fraction of an inch was saved in the layout. At the front axle, the crankshaft was the same distance from the top of the frame as on the original car, and the engine was set parallel to the frame. The transmission and axle housing was then just below the top of the frame, so that the bottom of the radiator was in line with the top of the frame. This took off 4 in., or 18 per cent, from the radiator height.

The radiator we use at the present time is just sufficient to cool the engine under the worst conditions encountered in the United States. We figured on taking off 2½ in. from the height of the radiator and adding 1½ in. to the sides. To obtain the same cooling area, it would be necessary to increase the present 3-in. depth of core to 5 in., and that increases the cost 42 per cent.

A layout showed that the rear of the cylinder-block would be 9 in. further to the rear than at present. To accommodate this additional length, a depression could be made in the dash. A depression of 3½ in. had already been made, and 9 in. more would bring this to

[13] M.S.A.E.—Editorial department, Society of Automotive Engineers, Inc., New York City.

[14] M.S.A.E.—Chief engineer, F. B. Stearns Co., Cleveland.

the junction of the toe board and floor board, assuming that the gear or chain housing was so designed that it did not interfere with the toe board. The housing would be approximately 13 in. wide, leaving about 12½ in. on each side for the driver's and passenger's feet.

If the prospective owners did not complain about this bump, everything would be okeh. We could then sit 1½ in. lower than at present and still have good vision. The front of the radiator could be 1½ in. back of the front axle, instead of 7½ in. ahead, giving a shorter hood. This has lowered the car, however, only 1½ in. Since the worm is over the worm-drive gear, 18 x 7-in. tires could be used instead of 20 x 7, yielding 1 in. more. As the radiator is closer to the driver, the driver could be 1 in. lower still without sacrificing vision. This means that the car could be lowered 3½ in., to 67½ over-all, unloaded.

If this procedure met with objections from the sales department or the purchasers, the only thing to do would be to lengthen the wheelbase 9 in. or abandon the eight-cylinder vertical engine. If the driver is pushed back 9 in., he must be elevated again at least 1 in. to have good vision. Therefore, with this type of car, a lowering of only 2½ in. is secured at the expense of an increase of 9 in. in the wheelbase. The solution then seems to be to abandon the vertical engine.

Driving Experience of an Engineer

E. WOOLER[15]:—The Timken Roller Bearing Co. obtained a front-wheel-drive car a couple of months ago for the purpose of becoming acquainted with the bearing problems, design and construction of this new type of drive. The car has now been driven by members of our engineering organization about 6000 miles, I personally having driven it about 2000 miles. Many of our trips were made in very wet weather. The car has also been driven in warm and cold weather, on snow and ice, and over as many kinds of roads as are available in the vicinity of Canton, Ohio.

E. WOOLER

I was very enthusiastic about the front-wheel drive at first, as my impressions were very good and I was able to get quite a new thrill out of motoring, principally on account of the car being capable of very high speeds. I have driven it at 97 m.p.h., according to the speedometer; and it can be driven around curves at 80 m.p.h., even on wet pavements, with a feeling of perfect safety, because of its good roadability. I drove from Toronto to Cleveland, a distance of 325 miles, all the way in a heavy rain, in less than 7 hr. However, I now believe that this same comfort, speed, and roadability could be obtained in a good rear-drive car if it could be built as low as a front-drive car. This, of course, would be rather difficult, on account of propeller-shaft clearance. No doubt the lowness reduces the frontal area sufficiently to increase the maximum speed considerably.

No trouble, such as excessive tire wear, wheel wabble and shimmy, has been experienced with the front-wheel-drive principle. It was thought that skidding tendencies were entirely eliminated, until a heavy fall of snow came on top of icy roads. We then found the skidding tendencies to be quite marked during deceleration; in fact, I skidded into a ditch while driving from Cleveland to Canton. This skid was caused by suddenly removing the foot from the accelerator while driving down hill on a curve at about 50 m.p.h. Before the brakes could be applied and the car got under control, it skidded into the ditch on the right-hand side of the road.

The rest of this particular trip was rather precarious, especially going down hill, as there was a distinct tendency for the rear end to get ahead of the front every time the foot was removed from the accelerator. Perhaps automatic declutching or a free-wheel arrangement in the drive would overcome this.

We have also had an opportunity to find out what happens in the case of a front-end smashup, as the car was accidentally driven into a brick post recently, bending the tubular front axle and the brake-drum and breaking the aluminum brake-shoe support. The damage to a rear-drive car under similar circumstances might possibly have been a broken radiator, fan and hood, at least equally expensive to repair.

Car Has Many Desirable Qualities

This particular car has a very attractive appearance, because of its low height. I was of the opinion that a car could be too low; but one soon gets used to this position and finds it exceedingly comfortable, particularly since the steering post has a very low rake and the wheel is in a good position in the driver's lap. The low hood and radiator permit good vision, even with the low seats.

The steering qualities are very good, but it might be quite difficult to get sufficient clearance for comfortable steering with a V-type eight-cylinder engine. The riding qualities of this front-drive car are very good indeed in the front seat, but only fair in the rear seat. I believe this to be due to the weight distribution. I feel that one of the biggest advantages of the front-wheel drive would be in connection with individual springing of all wheels, which has not been provided on any design produced in this Country, to my knowledge.

The engine in this particular car was rather noisy, possibly because of sacrificing quiet design to obtain power at high speeds. The fact that the vibration dampener is directly under the driver's feet is quite noticeable, as its operation can be felt during the periodic vibrations of the crankshaft. The mounting of the transmission and drive gears directly on the frame is no help in regard to noise, especially since these units are located at front end of the car.

The braking is excellent on this car, which may be attributed to the hydraulic operation and the extra weight on the front tires. The turning circle is very small, being only 36 ft., and the car will remain practically level while turning a circle at full lock with wide-open throttle, whereas a rear-drive car of the same class tipped over at about a 15-deg. angle and at lower speed. Several have questioned the cooling efficiency of this car because of the small frontal area of the radiator. I have had no opportunity to drive it in very hot weather.

[15] M.S.A.E.—Chief engineer, Timken Roller Bearing Co., Canton, Ohio.

Peculiar Characteristics Observed

Three rather peculiar characteristics of front-drive cars, which would not be likely to occur to any one without actual driving experience, are:

(1) A very high side wind, even amounting to a gale, does not affect the steering as it does that of a rear-drive car, especially one having disc wheels.

(2) The battery location is evidently quite a problem. The car with which we are experimenting had the battery under the hood, directly behind the fan. Its location has now been changed to a point outside the frame and under the right rear fender. The position under the hood made the battery so hot that an impression could be made in the pitch insulation with a finger nail; and the battery had to be of special design, with baffles, to prevent acid from being thrown over the engine from the blast of the fan.

(3) The most peculiar characteristic is that a front-drive car is much dirtier than a rear-drive car, because it runs into its own dirt and because of its lowness and the sweeping fenders, which are so much in vogue at present. The splash and dirt are thrown by the front driving wheels onto the rear window light, so that it is impossible to see out of the rear window after a very few minutes of driving in the rain. During one trip from Canton to Cleveland in the rain, it was absolutely necessary to stop and clean the driver's side of the windshield and the left front window three times, because a high west wind blew mud onto the left side of the car.

B. H. BLAIR[16]:—Does Mr. Muller think that the front-wheel drive is practicable for a light-weight, low-priced car? The front axle is far more expensive.

W. J. MULLER:—There is no reason why a front-wheel drive cannot be built at as low cost as a rear-wheel drive. It is a question of design.

Keeping Down the Length

MR. ENGLAND:—I had the opportunity of riding in and driving the original Ruxton car for virtually one day in New York City. The thing that interested me more than anything else was that it could certainly be driven around the corners faster than any rear-drive car that I have ever driven. This car could go around in a small or large circle, as fast as the engine would pull it at full throttle, without making the tires squeal. The car was remarkable in that respect.

At first I did not like the very low position of the drive, but I think I would become accustomed to it. What I should like to know is: how are we to find space for the engine and the passengers in a front-drive car? Mr. Muller has said that the front-wheel drive can be put in the same space as the rear-wheel drive.

MR. MULLER:—On our first front-wheel drive, we used a conventional transmission with a spiral-bevel unit attached to the front of it. The hood was 10 or 12 in. longer than for a rear drive, and the traction was not as good. Then we designed our present transmission, which embodies a centrally located worm drive. The constant-mesh gear, high-gear, clutch and second-speed gears are back of the worm, and the low and reverse gears are in front. Nearly all of the parts excepting the splined shaft are the same as in a conventional transmission, and the distance between the engine and the center of the differential was reduced 5 to 8 in. In this way we have been able to find room for the engine under a hood of normal length, and we moved the axle backward enough to secure the traction that was needed on very steep hills.

HOWARD DINGLE[17]:—Has Mr. Muller had any experiences such as were described by Mr. Wooler in driving on wet streets?

MR. MULLER:—I built a roadster with 114-in. wheelbase, for high-speed driving, and sent it to Indianapolis at a time when the mountains were well covered with snow. The driver reported when he came back that he had absolutely no difficulty in getting over the hills; the car handled very well and never at any time was it necessary to put on the tire chains which he carried.

R. P. LEWIS[18]:—Does the use of chains on the front wheels affect the steering?

MR. MULLER:—Absolutely not. We have driven a car with chains, under the rather unfavorable conditions of very cold weather and 35 lb. air-pressure in 6-in. tires, and they changed the comfort very little. Incidentally, the chains are much easier to put on.

MR. LEWIS:—Does the shifting of weight away from the power wheels during acceleration have any tendency to reduce the acceleration because of reduced traction?

MR. MULLER:—A low center of gravity is most important in a front-wheel-drive car. If the center of gravity were very high, the shifting of weight during acceleration or hill climbing would be considerable; but it is very slight if the center of gravity is as low as it can be in a front-drive car.

CHAIRMAN J. WEBB SAFFOLD[19]:—The rear axles of rear-wheel-drive cars bounce so much that the wheels are out of contact with the ground a surprising percentage of the time. Is not the tractive contact more constant with a front-wheel-drive car?

MR. MULLER:—The answer to that lies in the proportions of the springs and their loads. The conventional automobile has a rear spring about 5 ft. long and a front spring about 2½ ft. long; everything else being equal, the front wheel naturally is in contact with the ground during a larger proportion of the time than is the rear wheel.

Steering and Wheel Setting

MR. LEWIS:—What is the maximum steering angle that you believe is necessary?

MR. MULLER:—I should say that 34 deg. is the most that is ever needed. We are now using 32 deg., with 130-in. wheelbase, and our car will turn in a 38-ft. circle.

MR. ENGLAND:—Is there not a greater tendency for the front wheels to get out of alignment because of the torque on the wheels, resulting in more wear of the front tires?

MR. MULLER:—We have done considerable experimenting on that question and have not definitely determined whether it is best to have the wheels parallel, toed in or toed out. We have an experimental sedan that has been driven between 10,000 and 12,000 miles on one set of tires, and every angle of the tread markings is

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[16] M.S.A.E.—Chief engineer, bumper division, Eaton Axle & Spring Co., Cleveland.

[17] A.S.A.E.—President, Cleveland Worm & Gear Co., Cleveland.

[18] M.S.A.E.—Chief engineer, Salisbury Axle Co., Toledo, Ohio.

[19] A.S.A.E.—President, general manager, Saffold Engineering Laboratories, Cleveland.

still visible. We have been absolutely free from front-wheel shimmy.

MR. ENGLAND:—The experience to which I have referred confirms that; we tried the roughest streets we could find around New York City and turned corners at high speed with no evidence of wheel fight or shimmy.

GEORGE W. HARPER[20]:—Do you set the wheels with the customary caster?

MR. MULLER:—Yes, we are using 2 to 3½ deg. and have tried as high as 6-deg. caster. The camber is ½ in. in the diameter of the wheel.

QUESTION:—Is it more difficult to service your clutch than to service the clutch in a conventional car?

MR. MULLER:—We can remove the radiator and transmission and change the clutch in 1½ hr.

H. D. KINNEAR[21]:—My observation differs from that of Mr. Wooler in regard to the riding qualities in the rear seats of front-drive cars. I have driven both of the front-drive cars that are on the market and I believe that the rear seats of both of them ride better than the front seats. Regardless of the type of shock-absorber used, the lower unsprung weight at the rear does result in a better ride.

MR. ENGLAND:—What is the effect of the forward location of the mechanism on the noise; is it dissipated before it reaches the driver, or is it amplified?

MR. MULLER:—There is no reason for more noise in the driving compartment; it is a question of the insulation between the engine compartment and the driving compartment. When our first car was built, we found that half of the noise resulted from the lack of support at the front end of the transmission, so we added a cross member with a small rubber dampening device. That was more effective in reducing noise than all the other work that we did. I think the noise is less with the front-drive car.

A. A. NICHOLS[22]:—What does Mr. Muller make of the statement that the tendency of a front-drive car is to shiver when the driver's foot is removed from the accelerator? I have driven such a car for four years and have noticed no such tendency.

MR. MULLER:—I have had one such experience reported to me by an experienced driver but have never been able to find out from him just what happened, and I have never had such an experience myself.

Good Action on Road Reported

P. C. ACKERMAN[23]:—After the experience which Mr. Wooler has reported, we set out to see just how the front-drive car would behave on ice. We were able to drive the car up to speeds of 70 m.p.h.; and we found no tendency to skid during deceleration or when the brakes were applied, although the roads were well covered with ice.

As Mr. Wooler reported, we have found that the car covers itself with mud and dirt whenever the pavement is slightly wet. I think this to be a real disadvantage of that particular design. I also confirm Mr. Wooler's observation that the front seat rides better than the rear. We observe a very decided chopping action in the rear seat, which is entirely absent at the front; the car is extremely comfortable for the driver during long trips.

MR. KINNEAR:—We have been led to believe that it is necessary to drive a front-drive car differently than a rear-drive car. My first experience with actually driving a front-drive car was with the short-wheelbase car which Mr. Muller has mentioned, which I drove about 50 miles recently. I can see no difference in action between this and any other car; in fact, I liked its steering action exceptionally well.

Relative riding-qualities in the rear seat are a matter of opinion; they depend upon the springs and the relation of sprung to unsprung weight.

MR. ACKERMAN:—Designers of front-drive cars evidently have concentrated on the front-drive mechanism and have not given enough attention to points such as riding qualities and noise. The engine has extremely high performance characteristics, but it is noisy. No doubt the riding qualities also could be much improved by changes in the springs and shock-aborbers.

MR. MULLER:—As we had not very long for experimental work on the Ruxton car, we thought it best to use parts, as far as possible, with which we had had experience on rear-drive cars. We took the dimensions of a good-riding rear-drive car and adapted them to the front drive, and the riding qualities have certainly been improved by reducing the weight of the rear axle. There is danger of causing rattle in any front spring which transmits the driving action if the spring is too long, and we do not know how long it can be before such trouble comes. We are using 37-in. front springs with a deflection rate of 365 lb. per in. and 55-in. rear springs with a deflection rate of 135 lb. per in.

Mounting Affects Sensible Noise of a Unit

MR. ENGLAND:—Apparent noise does not depend so much upon the engine itself as upon the way the engine is connected with the other mechanism. We had experience with an engine which was practically noiseless in one chassis and was very noisy in another. All the vibration points in the engine came out very noticeably. We were obliged to put rubber mountings in several places, and still could never eliminate the noise. All the noise originated in the engine, and it was amplified in one car so much that it was unbearable.

E. G. BODEN[24]:—What was the difference in mounting between the noisy job and the quiet one?

MR. ENGLAND:—We adopted Belflex hangers at the rear end of the engine and a mounting entirely surrounded by rubber at the third point in front. This mounting was an improvement which made the job barely passable.

CHAIRMAN SAFFOLD:—Much work remains to be done on automobiles of all types toward damping out noise from the transmission, rear drive, carbureter and other units. I think the noises are magnified no more in a front-drive car than in a rear-drive car.

E. M. SCHULTHEIS[25]:—I do not agree with Mr. Kinnear that there is no difference in driving between front and rear-drive cars; a driver will unconsciously drive about 20 m.p.h. faster with a front-drive car over the same roads.

MR. MULLER:—Most drivers have an increased sense of safety in a front-drive car which leads them to drive

[20] M.S.A.E.—Chief engineer, Columbia Axle Co., Cleveland.

[21] A.S.A.E.—Sales manager of factory equipment, Gabriel Snubber Mfg. Co., Cleveland.

[22] Machinery design division, Goodyear Tire & Rubber Co., Akron, Ohio.

[23] M.S.A.E.—Assistant chief engineer, Timken Roller Bearing Co., Canton, Ohio.

[24] M.S.A.E.—Experimental engineer, Timken Roller Bearing Co., Canton, Ohio.

[25] M.S.A.E.—Automotive engineer, Timken Roller Bearing Co., Canton, Ohio.

faster than they have been accustomed to doing, without realizing it.

MR. BODEN:—How does a front-drive car behave when it gets into a rut, particularly an ice rut?

MR. MULLER:—A skid can be corrected much more quickly if the wheel that is skidding is a steering wheel than if it is a rear wheel, and it is much easier to correct a skid or to get out of a rut when the front wheels are doing the driving. This fact is very noticeable when a car has been driven off the side of a concrete road; it is hard to get back on the concrete with a rear-drive car and easy with a front-drive car.

J. H. SHOEMAKER[26]:—Has Mr. Muller noticed any difference in gasoline economy that could be ascribed to difference in traction with the front-drive car?

MR. MULLER:—Our records show that one of our five-passenger sedans has been driven between 5000 and 6000 miles with an average gasoline consumption of 13 miles per gal.

QUESTION:—Is there any difficulty in cooling the oil-pan of your engine?

MR. MULLER:—We have had some trouble from building up of the oil temperature, because the oil-pan is rather low.

MR. BODEN:—Is the deceleration in braking any more rapid in a front-drive car?

MR. MULLER:—There is no noticeable difference, although I have heard people say that there is.

Constant-Velocity Universal-Joints

MR. LEWIS:—Have you made any experiments with universal-joints which do not transmit uniform angular velocity?

MR. MULLER:—When we began building a car, no one was manufacturing the Weiss universal-joint and we had some difficulty before finding someone to make it. In the meantime, we tried to find some other universal-joint that would fill our requirements. One of the joints we tried has been used successfully on four-wheel-drive motor-trucks for a number of years. After installing this joint, I was accelerating while turning a corner when something pulled the steering-wheel out of my hand and I landed on the sidewalk. It seems that, when this joint had passed a certain angle, application of power tended to make the angle increase. However, we are now experimenting with another joint which has so far been entirely satisfactory. I believe it to be absolutely necessary that constant-velocity joints be used at each end of the floating shafts.

MR. NICHOLS:—The front-drive car which I mentioned is quite short. We used ordinary universal-joints in this for four years without any trouble; the lack of constant angular velocity is not noticeable on any ordinary curve of the road. The difference is apparent only at extreme angles. If the wheelbase is short, the turning angles can be kept low and no trouble will appear. Special joints would have been too expensive for our purposes.

QUESTION:—What has been Mr. Muller's experience in regard to wear of the steering-knuckle pivot?

MR. MULLER:—I believe we have less wear on these pivots than with a rear-drive car, probably because driving with the front wheels has a steadying effect on the wheel and because of the absence of wabbling.

Vertical Steering Pivots Advocated

MR. NICHOLS:—What is the structural difficulty in placing the steering-knuckle pivots where they belong, in the plane of the wheel? We did that with our car, and used no caster angle.

MR. MULLER:—When four-wheel brakes came in, I was connected with a company which did a lot of development work on the central-vertical-pivot axle, but we were never able to find any difference in action between an inclined pivot and a central vertical pivot.

MR. NICHOLS:—The difference is merely that the inclined pivot is subject to a constant side thrust, which gives a tendency to wear. The pivot pins on our car, which has been driven 22,000 miles, are absolutely vertical and are just as tight as they were originally. The car is made from a Ford engine and Ford parts for the front and rear axles to the extent of perhaps 80 per cent.

J. E. HACKER[27]:—Was Mr. Wooler's skidding experience due to one front wheel only being off the ice when he applied the brakes?

MR. ACKERMAN:—We tried to cause a skid by driving on a street where the tires on one side were on ice and the tires on the other were on dry pavement. Of course, it is possible to produce a skid in any car under those conditions; but we found that the front-drive car recovered extremely well from any kind of skid that we could produce and the skidding effect was not at all the same as with a rear-drive car.

R. E. FRIES[28]:—I look at the front-wheel drive as a fine product having extremely good possibilities. If it were in the same stage of development as the rear-drive car, many of the present criticisms could not be made. The most important thing to the public—who, after all, will be the final judge—is which design is the safer, and certain inherent features of the front-drive car make it safer. Some trouble has been experienced with regard to the caster and toe-in of the front wheels and other things of a minor character, including so-called front-end noise. It is a matter for the public to decide whether they want cars that are extremely low and safer, or cars that are as high as rear-drive cars usually are.

MR. BODEN:—Has Mr. Muller found trouble from dirt being thrown back from the front of the car?

MR. MULLER:—Our car has no running-board, only individual fenders with leather splash aprons, and it seems to throw noticeably more dirt from the front wheels. That may be because our front fenders are a little too short in front, as dirt is actually being thrown over the front of the front fender. We have heard it said that the car throws stones, but we have had no experience of the sort.

Rear-drive cars require rear fenders covering virtually the whole wheel, because the rear wheels throw so much mud that a car following closely will have its windshield badly spattered. The front-drive car throws very little dirt from the rear wheels.

THE DISCUSSION AT THE ANNUAL MEETING

ROSCOE C. HOFFMAN[29]:—As I see the front-wheel drive, after the studies I have made during the last 1½ years, while designing two different cars, the main

[26] A.S.A.E.—Sales manager, Gabriel Snubber Mfg. Co., Cleveland.
[27] White Motor Co., Cleveland.
[28] M.S.A.E.—Vice-president, Columbia Axle Co., Cleveland.
[29] M.S.A.E.—Parkstone Apartments, Detroit.

problem is that of getting the center of gravity of the powerplant as far forward as possible. A V-type engine and some sort of a short transmission, similar to that used in the Ruxton or one of two or three other types that are in existence, may be used to attain this result.

Most of the advantages of the front-wheel-drive car are of a secondary nature. It permits changes in car design which add beauty and novelty, resulting in considerable commercial value because of the sales appeal. Several types of construction may be used.

H. C. SNOW

Should conventional design be followed, the cost will probably be higher than that of the rear-drive cars until we have produced the front-wheel drive for several years. Eventually, the cost of building front-drive cars will be lower.

In the cars I am building, I am considering chiefly the questions of a short transmission, individually sprung wheels and individual wheel steering. However, individual steering does add to the expense.

HERBERT C. SNOW[30]:—One of the most outstanding advantages is the roadability of the car. Anybody who has driven one of these cars for any length of time admits, without question, that there is a noticeable difference in the performance of the car, the ease of handling, especially at the higher speeds and on curves, and the security of the car. The general design gives a low center of gravity, and many have claimed that this is the real reason for the roadability and ease of taking curves at high speeds. While we agree that this does have some advantage, the greater advantages come from the pulling effect of driving through the front wheels.

I do not exactly agree with Mr. Muller that the front-wheel-drive car should be designed around the universal-joint, although that is a more important unit than in the rear-drive car. I believe that everyone who has done any development work on front-wheel drives agrees that a constant-velocity joint is necessary, and it has been a real problem to develop a joint which is satisfactory.

Favorable Setting for Front Wheels

Conditions affecting the position of the steering-knuckle pivot are different on the front-drive car than on the rear-drive car. In front-drive cars, the propelling force comes from the front wheels and axle through the spring to the frame and in coasting this is reversed. In a rear-drive car, both driving and coasting forces come from the frame through the springs to the front axle and front wheels. A rearward angle of the pivots of 1½ to 2 deg. is customery on rear-drive cars. We have found that the best results are obtained with a front-drive car when the pivot is set vertical; it can even have a reverse caster angle of ½ deg. without causing road shock or shimmy. No angle is given to the pivot as viewed from the front, so turning the wheels causes no tendency to raise or lower the front of the car. We have come to the belief that such an angle is necessary in rear-drive cars to give the wheels a tendency to right themselves on a curve; but the front wheels of the Cord car have the tendency to right themselves and this construction makes steering easier.

Many have driven our front-drive cars over slippery ice, particularly at night, without realizing that the road was slippery until they tried to slow up, because they could not see its condition. We have also found that it is much easier to get a car of this type out of ruts in the snow, because the car immediately starts in the direction in which it is steered instead of being pushed along in the direction of the rut.

Another one of the main advantages of the front drive is the possibility of mounting the body lower and making the frame construction much stronger because the propeller-shaft and the rear axle do not interfere with it. The proportions of the hood and body seem to be pleasing, as judged by public reaction.

One objection brought up against the front drive is the long wheelbase. We have met this objection by a construction of the steering-knuckles which permits a much larger turning angle of the front wheels than is customary, so that our turning circle is smaller than on many rear-drive cars of shorter wheelbase. We have found no difficulty in regard to tractive effort.

W. R. STRICKLAND[31]:—I have driven only one front-drive car, and that went straight ahead on a slippery turn when I applied the power. I believe the claims are justified in connection with good, dry, clean roads, but not to the extent that the front-drive car will do anything that the rear-drive car will not. I think also that teaching drivers to handle a front-drive car will cause considerable trouble, as the difference is of a greater order than that between cars having different kinds of gearshift.

J. H. HUNT

G. L. MCCAIN[32]:—What happens when a car is being driven down a slippery road at a turn and the driver is not aware that the road is slippery?

MR. SNOW:—When the power is off, the condition is the same with either type of drive.

MR. MULLER:—There is a much better chance to keep the front wheels turning if they are the driven wheels; there is, therefore, a better chance of controlling the front-drive car than the rear-drive car.

J. H. HUNT[33]:—The engineering aspect of the front-wheel drive reduces itself to the question of whether the manufacturer can give his customers any more for a dollar with this construction than with the present designs. The only other reason for the new design will be if the general public decides that it wants front drives and is willing to pay for them. If that proves to be the case, every company will be making front drives

[30] M.S.A.E.—Chief engineer, Auburn Automobile Co., Auburn, Ind.

[31] M.S.A.E.—Assistant chief engineer, Cadillac Motor Car Co., Detroit.

[32] M.S.A.E.—Research engineer, Chrysler Corp., Detroit.

[33] M.S.A.E.—Patent section, General Motors Corp., Detroit.

within five years. I wish to congratulate these manufacturers who have had the courage to try out the public reaction on this.

Only a certain number of customers have the courage to buy a car embodying such a radical change in design, and a manufacturer having an output of 100,000 cars per year would not be warranted in such a venture.

W. G. WALL[34]:—After watching front-drive cars for several years, both on the race track and on the road, and doing some design work upon them, it is my opinion that we are at the beginning of an evolution of design in cars such as has not occurred for years. This may cause a material change in the form of passenger-cars. The great reduction in unsprung weight improves the riding qualities of the car very materially, and the position of the rear axle can be changed, if desired, in addition to the lower height possible for the car.

Engineers have yet to gain experience with front drive problems. I believe that the tendency will be toward engines and transmissions that will make it possible for the passengers to sit even further forward in the car than in the present conventional cars. One of the problems to be overcome is the increased need of quiet engines and gears. I believe that much progress will be made in front-drive cars and that many of them will be used. Front-drive cars will be better in many respects than our present cars, although they have some disadvantages as well as advantages.

W. G. WALL

HERBERT CHASE[35]:—I have been told that front-drive cars are subject to excessive wear of the front tires. This may be due either to the driving effort or to lack of ideal wheel alignment. I have noticed a wedge for adjusting the caster angle on the Cord front axle, which seems to indicate that the caster angle is delicate and needs provision for adjustment.

Effect of Drive on Tire Wire

JAMES E. HALE[36]:—Before the advent of balloon tires, tire troubles were confined to rear wheels. Balloon tires introduced troubles at the front, particularly on cars that were not carefully adjusted as to camber and toe-in and on tires that were not kept properly inflated. It was necessary to make definite recommendations that the wheels should be almost vertical and the toe-in very slight. Following these recommendations has minimized front-tire trouble. I believe that very careful attention must be given to this question on front-drive cars.

Car owners have come to expect anywhere from 14,000 to 20,000 miles of service from tires. I am afraid they will be disappointed in this with cars such as are now being built. Every advance in the performance of cars makes the tire conditions so much harder, and a mere reading of the advertising claims for current car models gives some idea of the additional work that tires will be called upon to do this season. If all this power is applied to the front wheels, a little mistake in the geometrical arrangement of the front-wheel mounting may cause very unsatisfactory tire service.

MR. McCAIN:—I should like to hear more of Mr. Muller's reasons for moving the center of the universal-joint away from the axis of the steering-knuckle spindle.

MR. MULLER:—We found the same stabilizing effect from moving the universal joint 1/4 in. in either direction from the spindle center.

MR. SNOW:—Mr. Hale has mentioned the desirability of the vertical steering-knuckle pivot and minimum toe-in for maximum tire life. The front drive has the advantage of good steering and driving conditions with the pivot quite vertical and a toe-in of 1/8 in. No inclination of the pivot is needed to avoid wander, as in rear-drive cars.

Our experience is that front tires have worn faster than rear tires since they have the driving effort to transmit; but the total wear of the four tires, if they are changed about from front to rear, is virtually the same with either drive.

The spring adjustment that Mr. Chase has mentioned is provided merely to take care of manufacturing variations in the axle and the length of the front springs, which makes correct caster impossible without adjustment. There has been no occasion for adjustment in service.

M. C. HORINE[37]:—We know that a deflated front tire is a source of great danger on a conventional car. Is it more or less dangerous on a front-drive car?

A. F. DENHAM[38]:—We have tried reducing the air pressure in a left front tire to between 3 and 5 lb. per sq. in. We drove the car on concrete and gravel roads up to rearly 60 m.p.h., and the car would hold the center of the road, hands off.

Worm-Drive Lubrication

QUESTION:—What has been Mr. Muller's experience as to the durability of the worm drive?

MR. MULLER:—We have had a little difficulty, mostly because of emulsification of the lubricating oil. We have found it necessary to use an oii containing 5 per cent of animal fat. The same oil without the animal fat would discolor the worm. We have also tried hypo lead compounds having a soap base, and they caused discoloration also.

CHAIRMAN T. J. LíTLE, JR.[39]:—Powdered lead carbonate ground in heavy straight-cut mineral-oil will maintain a film under conditions that no other lubricant will withstand. This compound has been known by machinists for a long time and is used on lathe centers. It is also used on small portable worm lifts which are frequently given excessive overloads, and it has been used in the steering-gear of a prominent high-priced passenger-car.

The lead carbonate will not settle out of the oil if it is carefully prepared. The carbonate is first mixed to a dense homogeneous paste and then gradually thinned out. If this is done too hastily, it will not remain fixed. Pure carbonate must be used; not the commercial grade, which contains lime.

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[34] M.S.A.E.—Consulting engineer, Indianapolis.

[35] M.S.A.E.—Associate editor, *American Machinist* and *Product Engineering*, McGraw-Hill Publishing Co., New York City.

[36] M.S.A.E.—Development department manager, Firestone Tire & Rubber Co., Akron, Ohio.

[37] M.S.A.E.—Sales promotion manager, International Motor Co., Long Island City, N. Y.

[38] Field editor, *Automotive Industries*, Chilton Class Journal Co., Detroit.

[39] M.S.A.E.—Engineer and industrialist, Detroit.

Heavy-Duty-Motorcoach Brake-Design

Discussion of the 1929 Transportation Meeting Paper[1] Presented by George A. Green

CONSTRUCTIVE criticism was prepared in advance by several authorities representative of different types of braking system, who presented their views at the meeting following the conclusion of Mr. Green's paper. Among the subjects they discussed are the merits of the combined booster and master cylinder in the vacuum system, the evolution of braking problems, the necessity for rigidity of brake hook-ups and the proper braking distribution between front and rear wheels. Emphasis is placed on the fact that, on pneumatic-tired commercial-vehicles, lack of space in which to provide for a brake of a size anywhere near comparable to existing passenger-car-brake standards constitutes one of the most difficult problems encountered in brake design.

Brake-problem difficulties are specified and emphasized, and the advantages of air brakes for heavy-duty work are stated. A further statement is that railway men can learn to operate and maintain other than air-actuated brakes on motorcoaches, because the problems are no different from those involved in the operation and maintenance of a passenger-car.

Among the other problems commented upon are brake pressures and drum scoring, the desirable amount of contact between brake-lining and brake-drum, whether front wheels or rear wheels shall lock first under braking pressure, and the sizes and types of tire which are most desirable. In conclusion, the discussion centers on whether or not brake-stopping ability can be computed accurately in advance of actual construction and trials for performance of a given braking system.

THE PREPARED DISCUSSION

GEORGE AINSWORTH[2]:—It is stated in Mr. Green's paper that "it can truthfully be said that the Pacific Coast pioneered the development of what is believed to be the most successful brake-operating medium that the heavy-duty-motorcoach industry has yet seen; namely, compressed air." No doubt this statement is founded on his personal experience and observation; but, had he said: "namely, compressed air and vacuum brakes," his statement would have met with my complete approbation. Since he has also stated that "the Westinghouse Air Brake Co. was the pioneer of the motorcoach air-brake system," I should like to state that the Bragg-Kliesrath Corp. is the pioneer in the development of vacuum brakes for motor-vehicle use, and my following comments with regard to vacuum brakes refer only to its product, hereinafter designated as the B-K vacuum brake. If the problem be one of "accelerating deceleration," by applying the B-K vacuum booster as a supplementary power-unit to any well-designed braking-system, either hydraulic or mechanical, it is possible to obtain a deceleration curve which will compare favorably with the best curves obtainable from air-brake deceleration tests. This assertion is based upon actual facts gleaned from tests in which it was my privilege to participate.

In connection with Mr. Green's enumeration of the merits and advantages of the air brake, the same favorable characteristics may prevail by the use of the vacuum brake as a power unit, together with even added advantages not obtained with air. For example, items (1) and (4) can be dismissed by saying that they are also characteristic of hydraulic brakes. Regarding item (2), when vacuum power is employed having a suitable ratio between power-unit and physical effort, the resultant pedal-pressure necessary to accomplish deceleration is usually less than one-third the physical effort that would be required without the vacuum booster.

In case of emergency, the full physical effort can be applied in addition to the power developed by the booster unit, because the use of vacuum as a power unit supplementing any braking system, either hydraulic or mechanical, means that the physical connection between the brake-pedal and brake-shoes is left intact and always available. This is an invaluable feature, as it gives the operator a combination of two separate systems which work in perfect harmony under normal circumstances; but, in the event of any possible failure of the power unit to function, the operator still has effective deceleration by increasing the physical effort.

Concerning the "definite sense of feel" mentioned in item (3), this is very decidedly present in the vacuum unit, and to my mind in a more progressive proportion than with compressed air. As to "the operation of doors, fare-registering devices and warning signals," item (5), these results also can be obtained by vacuum, and door engines operated by vacuum have given results equal to those given by air operation.

In regard to item (6), vacuum equipment is being used by many large-scale motorcoach operators, including public utilities companies, from coast to coast. Air-brake operation for motorcoaches can hardly in fairness be compared to that of railroads, as the principle of operation is quite different. While the air-brake system has its virtues, the legion of vacuum-brake users is growing and now includes a number of the leading commercial-vehicle makers of this continent, sev-

[1] The paper was published in the January, 1930, issue of the S. A. E. JOURNAL, beginning on p. 41. Mr. Green is a member of the Society and vice-president in charge of engineering for the General Motors Truck Corp., Pontiac, Mich. A synopsis of the discussion is printed herewith, but the abstract published with the paper is not reprinted.

[2] District manager, Bragg-Kliesrath Corp., Detroit.

TABLE 1—DECELERATION TESTS OF A 12,250-LB. VEHICLE FROM A SPEED OF 20 M.P.H.

Pedal Pressure, Lb.	Deceleration, Ft. per Sec. per Sec. Without Vacuum Auxiliary-Unit	With Vacuum Auxiliary-Unit
50	3	10
75	4	12
100	5	13
150	8	15
200	11	17
250	14	20
300	15	24

eral of whom have standardized on the vacuum-power brake after very thorough tests of both systems.

Vacuum-Brake Advantages Stated

The added advantageous features to be obtained with the vacuum system as compared with any other system are the much lower initial cost; the simplicity and fewer number of parts; the absence of operating costs, since vacuum is always present in the engine manifold at low throttle, in which position the engine is not called upon to do other work; the freedom from cold-weather troubles, such as condensation and freezing; and the very important advantage of retaining physical service-brake connections in case of an emergency. I wish to concur most heartily with Mr. Green in his forecast of the future development of heavy-duty braking-systems. Hydraulic operation, combined with a vacuum auxiliary power-unit, or some other system equally simple, inexpensive and efficient, is undoubtedly the way to ultimate brake design.

Combined Booster and Master Cylinder

In respect to the combination of the booster and master cylinder as an integral unit, illustrated in Fig. 9 of Mr. Green's paper, we do not favor this integral unit. The one illustrated was developed some three years ago, and we found the following disadvantages:

(1) It offers complications from a service viewpoint, because any repairs to the vacuum unit render the entire system inoperative, in addition to the necessity for bleeding hydraulic lines and the added time required for the removal and replacement of the unit. Electric-ignition manufacturers long ago learned that the combined generator-starter-distributor unit was entirely impracticable.

(2) The integral unit limits the advantage of leverage ratios between vacuum cylinder and hydraulic master-cylinder and, since each manufacturer's requirements are usually unique with his own product, it would mean the manufacture of a large number of different models to obtain desired results as to the relation between power and mechanical advantages.

(3) The separate units are lighter in weight and much more convenient for installation, and they permit greater liberties with respect to pedal ratios. Initial cost is also less, and servicing problems are minimized.

Evolution of Braking Problems

Mr. Green has ably set forth the future needs for efficient braking, all of which unquestionably point to the use of power. The braking problems of today are nothing more than the evolution of similar problems dating back long before the automobile, motor-truck or motorcoach were dreamed of. In early days the farmer who lived in hilly regions had his wagon equipped with a simple type of plain brake-shoe, which he forced against the wagon wheel by the power of a long lever placed alongside where he could easily grasp it with his hand when occasion demanded. Bracing himself in the wagon seat, he could easily exert the full power of his arms and shoulders and be assisted by the long leverage.

With the advent of the motor-vehicle it was soon discovered that both hands of the operator were required to control the vehicle; so, the long brake-lever of the wagon was cut off, bent to fit the available space and placed under the floor-board to be operated by the foot. With the lever thus shortened and the applied power or pedal force reduced, together with increased vehicle-loads and speeds, it became necessary for engineers to look about for mechanical contrivances of one sort or another to compensate for pedal effectiveness lost when levers were shortened.

Year after year loads have increased, speeds and powers have multiplied, but Nature's laws have remained unchanged. Inevitably, as braking requirements increase, the applied pedal-pressure must likewise increase and, since human strength is definitely limited, the application of power for braking purposes constitutes the only solution.

Table 1 shows the result of tests conducted on a vehicle weighing 12,250 lb. Deceleration readings were taken from a speed of 20 m.p.h. Table 2 shows the results of a tractor-trailer deceleration test from a speed of 20 m.p.h. The trailer was equipped with a vacuum brake-operating unit and had mechanical brakes on the trailer only; the servo-type mechanical brakes on the tractor were without power assistance.

Rigidity of Brake Hook-Up Essential

C. H. TAYLOR[3]:—One of the real problems of brake hook-up design is to obtain sufficient rigidity to prevent undue pedal-movement without corresponding brake effect. A considerable part of this elastic pedal-movement loss is due to bending deflections of the cross-shaft. To avoid this, cross-shafts are braced to cross-members and we find the deflection of the combination is still excessive. Development along the line of minimum elastic losses in the entire braking system is now in progress.

My experience with drum scoring seems to indicate that the unmachined or raw surface of a hot or a cold-pressed low-carbon-steel drum is almost unscorable; but the same drum is quite sensitive to such action after turning or grinding. It does not seem unreasonable to expect development along the line of improved final-

[3] Consulting engineer, Bendix Brake Co., South Bend, Ind.

TABLE 2—DECELERATION TESTS OF A TRACTOR-TRAILER UNIT FROM A SPEED OF 20 M.P.H.

The Servo-Type Mechanical Brakes on the Tractor Were without Power Assistance; the Mechanical Brakes on the Trailer Were Operated by a Vacuum Unit.

Pay Load, Tons	Gross Weight, Lb.	Pedal Pressure, Lb.	Deceleration, Ft. per Sec. per Sec. Tractor Only	Tractor and Trailer
0	17,400	170	11.6	20[a]
5	27,400	170	9.0	20[a]
10	37,400	170	7.0	15[a]
15	47,400	170	4.8	11

[a] The trailer wheels locked on full application of the vacuum brake; otherwise, the deceleration would have been better.

pressing methods to produce sufficiently accurate drums in the raw state to reduce or entirely avoid scoring troubles.

"Equalization" is a word which has, in my opinion, exerted a distorting influence on brake hook-up design because it has suggested the idea of adding specialized or artificial equalizing devices to the hook-up. The inherent unreliability of any completely equalized brake system does not seem to have been given due consideration. What I may term "artificial equalization," as opposed to the natural tendency toward equalization of a simple rigid system, presupposes equality in operating effort at all brakes; hence, if one brake control fails, all other brakes become inoperative. Fortunately, artificial equalization in mechanically operated brakes is rapidly disappearing, but it still exists in other systems.

Mr. Green mentions the geometry problems of the mechanical hook-up. These unquestionably exist with the classic rod-and-lever controls. While at times these problems are very difficult of solution they present one important possibility; that of enabling a transfer in braking effort to be made from rear to front brakes in proportion to tire load as the deceleration increases. With cable-and-conduit mechanical controls, no geometry problem exists.

Braking Distribution Discussed

The question of braking distribution between front and rear wheels suggests that the industry as a whole is still conservatively declining to take full advantage of front brakes. The bugbear of steering loss due to locking front wheels has inclined engineers to insist on rear wheels sliding before the front wheels slide, under all conditions. Superficially, this sounds logical, but my experience has entirely convinced me that it is questionable practice.

When rear wheels slide, they skid sidewise and there is no alternative but to follow the skid with the front wheels. If the skid is extreme, the car progresses sidewise or it spins, presenting a broad cross-section to strike any opposing object. With the front wheels locked, the car progresses in a straight line, irrespective of steering angle; but it immediately and accurately responds to the steering angle when the brakes are released enough to restore front-wheel rotation. Of the two alternatives, the latter is, I think, far preferable. I am told that this may be true for the skilled driver, but not for the general public. I answer that the general public is composed of a large majority of fairly skilled drivers, as is evidenced by the fact that so few accidents occur in spite of the vast car-mileage that is being rolled up each day.

The ideal brake mechanism is, in my opinion, one in which front wheels slide slightly before the rear wheels slide under all terrain conditions with the steering-gear in the straight-ahead position, and wherein the front brakes become rapidly less effective as the steering angle increases. Considerable prejudice may exist against this arrangement owing to lack of experience with it; but this, I think, will rapidly disappear with its use.

Vehicle Weights, Efficiency and Cost

As to the weight limit of a vehicle being 10,000 lb. for use with foot-controlled brakes without auxiliary power, I think this figure could be raised safely to 15,000 lb. without unduly crowding the limit. We have equipped coaches experimentally to 22,000 lb. with very favorable results, decelerations reaching 25 ft. per sec. per sec.

As to the efficiency of mechanical systems, this may be taken as about 70 per cent when the various principal bearings are lubricated. In this day of centralized chassis-lubrication, this does not present a major problem. Referring to the cost of the various systems, there seems little doubt that the several mechanical hook-ups are definitely lower in first cost than are any of the other competing systems.

In connection with the various feasible power brake-operating arrangements, one system has, I believe, been used successfully abroad, but it has had little or no development in this Country. It comprises the usual four-wheel brakes and a transmission brake. This latter comprises a resilient anchorage whereby the transmission-brake torque in either direction is used to apply the four-wheel brakes. By this arrangement practically unlimited power is available for applying the wheel brakes in a simple and direct manner. The hand brake in this case can operate directly on a cross-shaft to set either two or four brakes.

Anything effective along the line of taking action to prevent, or at least limit, the overloading of trucks would not only help solve the brake problem, but would decrease substantially the ton-mile cost to the user through its effect upon all units of the vehicle.

Space for Adequate Brakes Is Limited

H. D. HUKILL[4]:—Emphasis has been placed by Mr. Green on one of the most difficult factors encountered in brake design; namely, the lack of space on pneumatic-tired commercial-vehicles to provide for a brake of a size anywhere near comparable to existing passenger-car-brake standards. Some space around the drum for air circulation is essential, and the author arrives at a 22-in. base wheel which will permit good installation of a 17¼-in. internal expanding brake. But many other builders are maintaining a 20-in. wheel, which will necessitate a reduction in brake diameter, and 16½-in. brakes are now being considered for large high-speed vehicles; thus the problem grows.

Mr. Green's predictions for the motorcoach brakes of the future are interesting and worthy of very serious consideration as a guide for present development. There is still one other combination which he does not mention that seems to possess considerable merit; it comprises a central booster-unit—either vacuum, air, electric, or mechanical, as may best fit the existing conditions—operating internal expanding-shoe brakes at the wheels through a cable-and-conduit control-system. Such a combination fits into the picture ideally where an independent hand-operated propeller-shaft brake is provided. This combination also incorporates the various advantages listed by Mr. Green for air and hydraulic brakes, at the same time eliminating the disadvantages enumerated for such systems.

With air and vacuum boosters now being developed, it is possible to maintain a "fractional" control of the brake at the pedal and at the same time utilize this fractional control as a mechanical follow-up in case of failure of the power element. Such a control does not permit minimum stops in case of power failure, but is much preferable to complete inoperativeness of the pedal in case of failure.

Cable-and-conduit controls are adequate and entirely

[4] M.S.A.E.—Manager, power brake division, Bendix Brake Co., South Bend, Ind.

satisfactory for transmission of the forces involved, and have the advantage of eliminating geometrical problems. The question of friction loss in the conduit must be considered, but experience indicates that the efficiency of transmission of the cable-and-conduit system is equivalent to that of an hydraulic system when viscosity of fluid, packing-cup friction and side-wall thrust from the master-cylinder piston are taken into consideration.

Brake-Problem Difficulties Emphasized

S. JOHNSON, JR.[5]:—The keynote of motorcoach-brake design was struck by Mr. Green when he stated that motorcoach performance must be comparable with that of the average automobile, thus making the brake problem indeed difficult. When we stop to realize that street-car deceleration averages 4½ m.p.h. per sec. and is accepted in many cases as the maximum that can be tolerated without discomfort to the passengers, we would expect the deceleration of a motorcoach to be on a comparative basis. But this is not true because many automobiles decelerate at very much higher rates when their brakes are in good condition, especially in an emergency and, since motorcoaches travel in company with automobile traffic, they are called upon to give the same or as nearly equal performance as is possible. Automobile traffic is so dense and cross-streets and cross-roads are so prevalent that emergencies arise frequently; because of them a reserve of power must be provided in any brake system.

The many years of precedent that lie behind the development of the air-compressor, upon the reliability of which depends the success of the entire air-brake system, are referred to by Mr. Green and some points in connection with its design are interesting. It was necessary that it should operate successfully at high speeds and have adaptability so that it could be employed with as many existing engines as possible. To be in line with the engine-manufacturers' practice of keeping auxiliaries as small as possible, it was necessarily designed for compactness as well as for quietness of operation and efficiency over a wide range of speeds. Further, it must possess interchangeability as a unit or in detail so that, if required, a complete unit can be replaced quickly or parts repairs effected easily and economically. The compact form of compressor designed to meet these requirements provides adaptability by including means for operation from either end. The drive may be direct from an auxiliary shaft; through gears; pulley and belt; or chain and sprocket; or, as more recently practised, attached directly to the engine crankshaft.

The necessity for high speeds, quietness and efficient operation led to the adoption of the rotary-type mechanically operated inlet-valve driven positively by rotation of the inlet-valve shaft which is operated from the crankshaft through helical gears integral on each of the two shafts. Many problems were encountered with lubrication and the force-feed system was finally adopted. The original design has been subject to the improvements required by increased severity of duty. One of the latest improvements is the change from the inverted type of mushroom discharge-valve originally employed to a disc valve that was developed for this purpose. The latter type is superior to the mushroom type in that it has less inertia during operation and, since there is little tendency for it to pound, the disc valve is somewhat quieter and increased life is secured.

The definite sense of "feel" referred to in item (3) of the listed advantages of the air brake is made possible by the self-lapping feature of the brake valve. Graduated-release operation of the brake is essentially important, and the brake valve employed in the Westinghouse system can release the brake in pressure gradations of 1 lb. This feature makes it possible to eliminate the so-called "fanning" of the brake, because maximum pressure can be transmitted to the brake chambers at the highest speed and then reduced gradually as the speed decreases until, at the end of the stop, theoretically there should be no air left in the brake chambers, or at most only enough to hold the coach at a standstill. In every case where this method of braking has been demonstrated to the drivers, we have noticed immediate improvements in their performance.

Other Features of Air Brakes

Referring to item (1) of the disadvantages of air brakes, a laboratory cannot set up all the varied conditions that a new development will encounter in actual service. We have 12,000 air-brake equipments in service and, with reasonable maintenance and inspection, they operate over an 18-month period before needing an overhaul. The tubing failures and breakage referred to in item (2) have been so infrequent as to be negligible. Expansion or vibration coils in the tubing are used where necessary, and a liberal use of clamps is recommended. Naturally, the coach builder, as well as the air-brake manufacturer, was obliged to learn the vital vibration-points on a particular chassis. To obviate the possibility that the driver will be uninformed regarding the air pressure in the system at any time, gages are conspicuously installed on the dashboard.

Regarding item (3), it would be interesting to compare the air-brake system with say a mechanical-brake system plus some kind of amplifier. With the elimination of cross-shafts and levers, together with the savings that are effected in manufacturing or assembling the chassis with tubes instead of rods and levers, it is probable that the increase in cost resulting from the substitution of air brakes would not be excessive. Any increase involved would be warranted in the cause of safety alone.

The universal acceptance and adoption of the air-brake system as standard equipment on the larger motorcoaches and trucks by the leading manufacturers, and the considerable interest being displayed at present by some of them toward air brakes for smaller motorcoaches and trucks, it probably is safe to predict lower costs as a result of increased production.

Concerning item (4), the design of the air-brake apparatus was made as simple as possible in view of the functions which it is called upon to perform. We find that, with a reasonable amount of individual instruction and a judicious distribution of instruction pamphlets by the air-brake service-representatives, the mechanics who service the motorcoaches soon become entirely familiar with the system.

Motorcoach Control Must Be Effective

To be able to control a motorcoach and bring it quickly and safely to a stop is a more important element in its operation than is that of imparting motion, for economic reasons as well as for reasons of safety

[5] General engineer, automotive brake division, Westinghouse Air Brake Co., Pittsburgh, Pa.

and convenience. If a rapidly moving vehicle cannot be stopped easily and quickly, it is impossible to profit from the obvious advantages of the potentially higher speeds, a principle early recognized and acted upon in connection with railway transportation.

The degree of safety in the operation of automotive vehicles, the same as that for railroad vehicles, and the close adherence to the schedules which must be maintained, are very largely dependent upon the effectiveness of the brake-control available. It therefore follows that the solution of effective control for motorcoaches operating at prevailing speeds and under a great variety of conditions of weight, grade and road surface, obviously lies in a power-brake system which is capable of developing sufficient braking force and which is flexible, that is, having sufficient available force so that a loaded motorcoach can fully utilize the high adhesion of dry roads while operating at high speeds, with provision for easy adjustment to suit the more favorable conditions of lighter weight and slower speeds. The air-brake meets all of these requirements. It has been associated with transportation by rail for more than a half a century, and has played an important part in the development of steam and electric railroads. We feel now that it is doing its share in the development of motorcoach transportation.

Both Types of Cylinder Used

H. C. BOWEN[6]:—The idea expressed by Mr. Green that the railroads now operating motorcoaches should use air brakes on them, because they are familiar with air brakes as applied to railway equipment and understand and appreciate the problems of service involved, does not, in my opinion, justify the thought that the air brake is the most suitable for motorcoaches. Where air brakes are used on motorcoaches they are not exactly the same as those used on railroad trains and the service problems are vastly different. In my opinion, the railway man can learn to operate and maintain the hydraulic brake on motorcoaches, since the problems are no different from those involved in the operation and maintenance of hydraulic brakes on his passenger-car, with which he undoubtedly is thoroughly familiar.

For heavy-duty motorcoaches, we use both the external and the internal cylinder because both have advantages. The external cylinder can be replaced easily and can be serviced when necessary without the removal of a wheel or without jacking it up; further, it is not exposed to the drum temperatures.

We are using internal wheel cylinders up to 2 in. in diameter, and putting a load on the brake-shoe as high as 3000 lb. Those cylinders are very successful, and we find that they will withstand such heat as the wheel bearings will withstand. We have run them until the grease has fried out of the wheel bearings, and still the wheel cylinder has functioned.

When designed for the 20-in. wheel and dual tires, any type of brake should be ventilated, for the good of the brake-lining and for that of the wheel bearings. There is no mechanical loss with the cylinder mounted between the brake-shoes directly, and this is an advantage. When the cylinder is mounted outside, the cam and lever must be employed, and there is a certain loss of efficiency due to the cam and lever which does not occur when the cylinder is interposed between the two brake-shoes.

When designing a brake we are compelled to lay it out to fit the wheel diameter the manufacturer specifies; that is, the 20-in. wheel. The largest brake-drum we can use is of $17\frac{1}{4}$-in. diameter. The same statement applies to the lighter trucks on which 16-in. drums are used; that is, we lay out the brake for a certain weight of truck and a certain size of wheel.

Brake Pressures and Drum Scoring

After having attended many dealers' conventions I notice that, with regard to the bringing out of a new model, the first question from the dealer and all the motor-truck and motorcoach operators is, "Can I install larger tires?" Both the dealer and the user want to install tires that will carry excessive loads; but they do not realize that when a 2-in.-larger wheel is installed it is practically the same as installing a 2-in.-smaller brake. When those larger tires are used, the operator has trouble on account of drum scoring and brake-lining wear.

It is a very simple matter to check the pressure that is exerted on the brake-shoes of hydraulically operated brakes. On our $16\frac{1}{2}$-in. brake that is $2\frac{1}{4}$ in. wide, we use pressures on the ends of the shoes as high as 1700 and 2000 lb. per sq. in.; for the 4-in. brake, about 2500 lb., and for the 5-in. brake, as high as 3100 lb. We find that those pressures work out satisfactorily.

All standard brake-linings will score the brake-drums if the pressure is high enough, and all will work satisfactorily if the pressure per square inch on the brake-lining is kept low enough. But when tires of larger diameter are installed, it is necessary to increase the pressure on the brake-lining to get braking force enough to stop the vehicle, and this causes trouble.

THE ORAL DISCUSSION

A. J. SCAIFE[7]:—Regarding the advisability of using the full 360-deg. contact between brake-lining and brake-drum, the railroad companies experimented years ago with longer brake-shoes that covered more of the car-wheel tread than do the present-day short brake-shoes. But the longer shoes did not cool well because they covered too much surface and the heat dissipation was poor. Would not that effect apply also to 360-deg. contact for motor-vehicle brakes? Would it not be better to use wider brake-shoes without covering the entire brake-drum, and thus provide better circulation of air?

G. B. INGERSOLL[8]:—Complete 360-deg. contact cannot be obtained, but I believe it can be approximated closely. It is impossible to get full contact-efficiency at the extreme points of the shoe. On our experimental truck, the single brake-shoe was mounted on a chassis that had a maximum service-capacity as high as 15,000 lb. The single shoe was equipped with the ordinary folded and stitched lining and, to date, it has operated successfully up to nearly 25,000 miles. The state of that lining today leads me to believe that the truck will still operate successfully on that type of lining for

[6] Engineer, Hydraulic Brake Co., Detroit, Mich.

[7] M.S.A.E.—Consulting field engineer, White Motor Co., Cleveland.

[8] M.S.A.E.—Chief engineer, Federal Motor Truck Co., Detroit.

50,000 or possibly 75,000 miles without replacement. I believe that the secret of this success as regards the degree of contact, length of life and efficiency, is dependent upon a type of shoe having a channel section with increasingly high side-walls which maintain a practically perfect circle in both the closed and in the expanded position so that equal pressures prevail throughout the entire contact surface of the drum. That the practically 360-deg. contact will be successful seems to be borne out by the experience we have had so far with that type of brake.

MR. AINSWORTH:—We all know that the total power developed and applied to the braking surface will result in a unit pressure per square inch that is commensurate with the area of the lining that engages the drum. Where the area of the lining engaging the drum is such that minimum unit-pressure is exerted and then increased proportionally under application pressures, the temperatures run up so high as to eliminate almost entirely the coefficient of friction between the lining and the drum, which renders the brake increasingly inefficient under sustained application pressures. Therefore, the full contact-area between the drum and the shoe is a very desirable factor, because it results in a consequently lower unit-pressure due to the increase in lining contact-area, which tends to prolong the life of the brake and to maintain a higher standard of deceleration, even under the most extreme conditions.

Locking of Front Wheels Undesirable?

MR. SCAIFE:—I cannot agree with Mr. Taylor that the locking of front wheels is preferable to having the rear wheels lock after brake application. As I was making a turn on a slippery asphalt pavement in a car equipped with front-wheel brakes some years ago, the front wheels locked and the car went straight ahead into a telegraph pole. This may have been because of my inexperience as a driver at that time. But recently, while driving a passenger-car equipped with hydraulic four-wheel brakes which had just been inspected and adjusted so that the brake clearance on the front wheels was very close and the front brakes were applied first, I applied the brakes on a highway that was muddy to avoid bumping a car ahead which had stopped suddenly and I went into the ditch because the front-wheel brakes locked the wheels and I could not turn. I changed conditions later so that the rear-wheel brakes were applied first and, under exactly the same conditions on the same highway while driving personally, I had no repetition of my former experience.

I can see no need for controversy about these different types of brake. They all have their field of application. I agree with Mr. Green that, without question, air brakes are best for heavy-duty motorcoaches and heavy-duty trucks, and our company uses all types of brake. On the lighter trucks the hydraulic brakes are very satisfactory, and on our medium-weight vehicles the hydraulic system equipped with a booster unit is giving good satisfaction. Each of these different types of brake equipment has a decided field of its own.

Sizes and Types of Tire Desirable

A. W. SCARRATT[9]:—Regarding more effective means for the ventilation of brake-drums, that is a matter of no small concern to the tire manufacturers as well as to the engineers and the motor-vehicle operators. It has been recommended that serious consideration be given the using of the 22-in. series of tires and rims for heavy-duty trucks to improve heat-dissipation conditions with reference to brakes. When will a complete series of 22-in. tires and rims be available?

G. M. SPROWLS[10]:—We already have available a complete series of 22-in. balloon tires and rims. The high-pressure tire is disappearing rapidly at present. There is an increasing demand for balloon tires. We could easily make the 22-in. high-pressure tires if there were a demand, but I believe there will not be much demand for them. On both motor-truck and motorcoach service we find that the balloon tire is giving very good service, especially in long-distance work.

It would be very desirable if everyone interested would standardize on one rim-diameter and, at least from a tire viewpoint, I would favor the 22-in.-diameter rim and the elimination of both the 20-in. and the 24-in. diameter rims.

MR. SCARRATT:—You have your regular high-pressure tire; then you bring out a super high-pressure tire for dump-truck and other heavy work. So, with several varieties of high-pressure tires available now to meet certain conditions in which you evidently have felt that the high-pressure tire was the right tire to use, how is the balloon tire going to supplant the various high-pressure tires and give the satisfaction in service that they were intended to give?

MR. SPROWLS:—We must give the customers what they want. If they demand high-pressure tires, we will make high-pressure tires; if they want balloon tires, we will make balloon tires. Even in dump-truck service we have found that, in many cases, balloon tires are more desirable than high-pressure tires. I would not be surprised if the balloon tire is not soon used more in dump-truck service than is the high-pressure tire.

J. W. SHIELDS[11]:—Regarding the size of rim that is to be used, we fully appreciate the problem that the designing engineer of motor-trucks and motorcoaches faces when the public or his sales organization demands the use of a small-diameter wheel. I believe that we can safely pin the blame directly on public demand rather than on the engineering organization, in the majority of the cases. The "buck" cannot be passed back to the tire organization because, if I am any judge of the opinion and the attitude of the tire industry, I believe we almost universally agree that the tire of larger rim-diameter is preferable; in other words, a 24-in. or a 22-in. rim is far preferable, so far as tire service is concerned, to a 20-in. rim. Do you agree with that, Mr. Sprowls?

MR. SPROWLS:—Yes.

MR. SHIELDS:—We express it by saying that if the diameter of the rim is reduced 1 in., the tire service is reduced by approximately 10 per cent. In other words, a 20-in.-rim tire will give 20 per cent less service than a corresponding 22-in. tire of the same construction and cross-section, operated under the same service conditions; and we can expect almost a 40-per cent difference in service life between the 20-in. and the 24-in.-rim tires. It is therefore logical that the tire industry would tend to favor the larger-diameter tires, not only because of the greater service, but because they also minimize difficulties caused by brake heating.

[9] M.S.A.E.—Chief engineer, motor-truck and coaches, International Harvester Co., Chicago.

[10] M.S.A.E.—Manager, highway transportation department, Goodyear Tire & Rubber Co., Akron, Ohio.

[11] M.S.A.E.—Sales engineer, Firestone Tire & Rubber Co., Akron, Ohio.

MR. AINSWORTH:—Someone has said that if the 22-in. rim became standard, some brake manufacturer would immediately increase the diameter of the brake-drum by 2 in. and thereby eliminate the advantage of increased dissipation of heat. So I imagine that the effect would be that more brake-lining would be supplied by the manufacturer to accomplish the same rate of deceleration, and that the same overheated conditions would have to be endured by the tire manufacturers and the operators.

Can Brake-Stopping Ability Be Computed?

W. G. RETZLAFF[12]:—The stopping of a 120-hp. tractor-trailer carrying say a 20-ton payload is an entirely different proposition from stopping say a 25-passenger motorcoach; some of the drivers travel at a speed of 25 to 40 m.p.h. Our company is constantly assailed by salesmen who represent the different types of braking mechanism and claim that theirs is entirely adequate for our purpose; but no one has given us any figures stating just how fast a gross load of 1000 lb. can be stopped from a given road-speed when using the braking system the salesman represents. The salesmen place the entire responsibility upon the operator, regarding the deceleration rate for a given load at a given speed for a specified braking system, and our engineers must choose by rule of thumb what type is best suited to our needs. On this basis we use the Bragg-Kliesrath vacuum brake for vehicles that carry a payload not exceeding 10 tons; for larger payloads, we recommend Westinghouse air-brakes.

It is true that deceleration-test figures are available, but they do not take into consideration the frequency of brake adjustment that is necessary. It also is true that a given pressure can be applied through an excessive brake-leverage and that a heavy load can be stopped thereby, but how many stops can be made before brake adjustment is again necessary? The number of effective stops that can be made before brake adjustment is necessary should also be considered, because it is too much to ask operators' employes to crawl underneath the vehicles several times per day to make brake adjustments because the power at the operating end of the braking system is not great enough to stop the load.

MR. AINSWORTH:—It seems that Mr. Retzlaff refers to the deceleration-test figures cited in Tables 1 and 2 of my previous discussion. In determining these figures the actual tests covered a period of three weeks of constant running in, testing and re-testing. Prior to the tests the vehicle was run daily for four days down 11-per cent grades and 7-per cent grades and on the straightaway under continual applications of pressure that were largely of an emergency nature, and only one brake adjustment was necessary.

I think Mr. Retzlaff's point would be well taken if the power device used were not sufficiently provided with reserve travel of the cylinder, the rods and the levers to compensate for wear; but, on these tests, the brake cylinder had ample capacity, the deceleration characteristics were constant, and one application after another was made with but little variation in the rates of deceleration. One-half the cylinder travel was still available to compensate for wear at the conclusion of the tests. For ordinary rates of application under average operating-service conditions the system would be capable of operating the trailer without necessity for adjustment for at least 10,000 miles.

MR. BOWEN:—Many engineers are capable of designing a braking system that will stop a vehicle within a given distance after assuming a certain coefficient of friction, a certain tire-diameter and a certain load. Then they try it out; if it is inadequate, a more powerful system is installed and, if the vehicle is over-braked, a less powerful system is used. I think it will be several years before we can design brakes that will perform as they are designed to perform.

Several factors enter into the performance of a braking system that are indeterminate and that prevent accurate prediction of what its performance will be. Differences of opinion also exist as to how the brake-system should function. Only lately would engineers admit that the width of a brake-shoe has to do with anything other than brake-lining wear; yet we know that a shoe of equal diameter but of greater width will give greater torque and a quicker stop for the same input of force on the ends of the shoe. In operation, at no two stops is the brake-drum likely to be at the same temperature. Higher drum-temperatures mean greater drum-diameter, and the brake-shoe must either change its shape to fit the drum during its variations in diameter or the brake-lining wears more at one point of the shoe than it does at another.

MR. JOHNSON, JR.:—I concur with Mr. Bowen and could cite instances in connection with railroad-train braking that bear out his statements. The ratio between the force developed in the brake cylinder and the force delivered to the brake-shoes at the rim of the wheel has been practically standardized for more than 45 years, yet could enumerate for an hour the data that must be known before one could even calculate theoretically how quickly a given railroad train could be stopped; and the same requirement applies to automotive vehicles.

[12] M.S.A.E.—Transportation engineer, Fruehauf Trailer Co., Detroit.

Airplane-Engine Development and Operating Reliability

Roland Chilton[1]

St. Louis Aeronautic Meeting Paper. Illustrated with Photographs

SO long as airplanes land at about railroad-train speed and traverse the average scenery, reliability must be the prime consideration in designing their powerplants. Safety is the factor in air travel about which the public is doubtful, and the required reduction in accidents must include the small percentage attributable to engine failure.

The author emphasizes that practically all parts failures in engines are from fatigue originating at small local defects in the material or from resonant vibrations which carry the stresses beyond the safe fatigue-limit of the material. He states that complete development of the electro-magnetic or equivalent inspection methods for detecting minute local imperfections will eliminate first-class failure. The importance of determining that no natural resonant-vibration periods of the parts are encountered within the operating speed range is emphasized.

It is also pointed out that reliability cannot be demonstrated by tests on a few sample engines, but that a large number of units must be observed in extended service to demonstrate that the factory inspection-system is holding the materials and fabrication within the required limits. The author asks a number of questions of the airplane experts directed at improved installation and increased over-all efficiency of the complete powerplant and airplane.

One feature commented upon in the discussion is to the effect that the aircraft industry is new and its production capacity is about five times larger than the market. One of the best ways to create a market is to reduce costs, it is said, which necessitates finding out what is going on and how to eliminate waste effort and additional costs of development. Questions concerning heat dissipation from valves, large-diameter valve-stems and the like are asked and answered. A statement is also made by a discusser that it is possible, with a given fuel, to increase the power-output of the engine if a sacrifice is made in fuel consumption, but that is the reason it is so important for the airplane manufacturers to specify what they want.

THE TYPES which become standardized during the development of any new industry follow the styles with which the initial financial successes were made, rather than technical considerations as to the merits of the various designs available. More new capital is applied to production of new makes along accepted lines than in developing new types of greater potential performance. The late abnormal confidence of the investing public in aircraft projects increased the number of manufacturers without developing any radical types to the production stage.

So long as airplanes land at about railroad-train speed and traverse the average scenery, reliability must be the prime consideration in designing their powerplants, at least as regards failures involving forced landings. The required high order of reliability cannot be achieved by the mere elimination of such defects as are developed by dynamometer endurance-tests on a few sample engines. For example, if less than 1 per cent of engine failures per 1000 hr. be aimed at, at least 100 engines must each be operated for that length of time to demonstrate that the inspection system is keeping the variations in material and manufacture within the safety factors and tolerances afforded by the design. Extended operation of a substantial number of units is the only practical criterion; hence, a pressure is exerted on engineers to adhere to time-proved details. A few types of aircraft engine may have reached the "experience figure" cited, but every popular automobile model has exceeded it many hundreds of times. Further, the power-weight ratio and the cruising-to-maximum-power ratio make aircraft-engine service many times more severe than that for road vehicles; hence, at its normal low average-output, the automobile engine is as yet the more reliable of the two. It is encouraging to reflect that, in spite of the great discrepancy in their specific weights, the airplane engine is already probably more reliable at high output than its forerunner, while every service failure corrected by a change in design is a step nearer to the time when the airplane engine will also be able to concede, in a reliability comparison, its handicap of high power-factor.

Press Reports Should Be Accurate

In one sense the situation develops in a circle. Improvements in reliability grow out of increased use, while any large increase in aircraft use probably awaits the demonstration of improved reliability. This com-

[1] M.S.A.E.—Consulting engineer, Wright Aeronautical Corp., Paterson, N. J.

ment is based on my conviction that safety is the factor in which air travel is popularly regarded as doubtful and that, until landing characteristics of airplanes are drastically changed, engine failures on transport machines are incompatible with public confidence. For every prospect lost because of high cost, there are many who could afford to fly but would not if you paid them, because it does not seem safe after reading the press accounts.

After newspaper correspondents discover that breakage of the landing-gear in a bad landing usually puts a plane on its nose, and that this is not caused by "suddenly nose-diving into the ground" from normal flight, as they usually report, this condition will be improved. So long as airplanes are likely to spin if stalled, and while some planes present difficulties in recovery if the spin is allowed to develop, it is only fair to explain to the gentlemen of the press that the stall and spin are like the violent skidding of an automobile, the former being bad business at low altitude, and the latter at high speed near the usual obstructions. The difference is that automobiles on greasy surfaces may go out of control without warning, whereas no approved airplane will stall itself. Nothing could persuade a competent operator of either to initiate these maneuvers deliberately except under the special conditions necessary for safety.

Newspaper men would be able to understand this information if we took the trouble to pass it along. It is not reasonable to expect a man who has to cover all earthly occurrences to have a technical knowledge of each. If journalists knew the facts they probably would agree that the "news value" of an airplane accident is not destroyed by mentioning that it originated in the pilot's error and not in an unprovoked and sudden nose dive to which any airplane is liable at any moment. As Elliott White Springs has written, "the public regards an airplane as requiring a pilot who is a cross between a trapeze performer and a sword swallower and, every time one of them cracks up, 119 million readers of the press headlines exclaim 'I told you so.' "

Necessity for Greater Safety Stressed

It is thought that when the airplane comes to be generally regarded by those outside of the industry as comparable in safety with the automobile, the added costs of air travel will not be found to prevent its reasonably rapid expansion. But no compromise with safety will be tolerated, because the airplane still suffers by being relatively new and the popular mind shies from the unusual, concerning risk as well as other things. The average person instinctively feels that it must be much more disastrous for an airplane to hit a tree than for an automobile to hit one, but this is a conclusion with which the author, having experienced both, emphatically disagrees. The 40,000 annual deaths due to automobiles do not weigh upon the minds of automobile users, and the development of the airplane to corresponding popularity will be witnessed by a succeeding generation which will be brought up to regard airplane accidents as "news" similar to that of automobile accidents and not, as at present, as being in an entirely different category.

It would be presumptuous to attempt to predict in terms of actual structure the outcome of the next few years of development in aircraft powerplants, but I do not hesitate to state that the recent optimism as to the rate of increase in aircraft use will not be fulfilled unless these developments result in increased reliability.

Danger as Related to Engine Stoppage

A portion of the general public may now be aware that an airplane does not necessarily fall to the ground immediately its engine stops; but most of them know that a landing, usually on unprepared ground, is the necessary sequel. Many prospective passengers already know that a dead-stick landing cannot be slowed up, delayed or shortened to suit unfavorable terrain. It is all very well to explain that from an altitude of several thousand feet the pilot has several minutes to choose his field; but this does not impress the observant man who has flown over districts which apparently afforded no choice of landings, particularly as he may be aware that we have not yet incorporated a "glide stretcher" on our standard planes, although young pilots may have been told, jokingly, that it constitutes an important accessory.

It is also true that, with a competent pilot and good luck as to when the engine failure occurs, a forced landing is unlikely to result in personal injury; but it is still an adventure which most men would not incur except for some special urgency. Further, the average passenger is not aware of what constitute the actual risks of flying and has merely a general idea that a machine high in the air with no visible means of support is likely to fall at any minute, yet we may truthfully assure him that all the dangers which he can specify are imaginary. Engine failure is responsible for but a small part of all fatal aircraft accidents, but none of these facts disprove the statement that engine failure adds to the actual and to the supposed risks of flying.

The multi-engined craft may greatly reduce the chance of complete engine-failure but they sometimes overtax the pilot, as when a wing engine cuts out on the take-off or in a steep bank, or when the plane is near the stall point. It seems that a very short reaction time is then given the pilot to realize on which side the engine has cut out and to make the appropriate readjustments. If there is a system, such as offset vertical fins, which will compensate automatically for this condition, many airplane designers seem to have overlooked it.

In spite of these conditions airplane designs are tending toward increased performance at the expense of increased landing-speed and, since the growth of air-transport depends primarily on public confidence, reliability must be insisted on as taking precedence over all other virtues in powerplants which might compromise it. Reliability is not always advanced by new developments which may effect improved performance, but is secured by the gradual correction of all design details which are found subject to failure in service, and by maintaining continuously a high order of uniformity in materials and manufacture. With a sufficient number of units under observation in extended service, it is not difficult to eliminate failures due to design.

Detection of Imperfect Materials

The elimination of fatigue failures caused by local imperfections in material is more difficult, but recent developments in inspection methods, such as the "magnetic induction analyzer," are rapidly overcoming these difficulties. This machine detects hair cracks, inclusions, segregation, and other defects in bar stock, which is fed rapidly through induction coils, and incidentally

excites in anyone who watches it a decided preference for engines built from stock so selected.

From the viewpoint of utility, the engine and the airplane are interdependent units. The engine designer cannot achieve an efficient product unless informed on the many airplane requirements. I have seen many installations which indicated that the plane designer lacked the reciprocal information. These remarks are intended to spur the plane experts to come forward with some criticisms of existing powerplants as to their shortcomings in increasing the efficiency of the completed craft. If the airplane designers will tell us where our engines fail to fit in front, or elsewhere, on a well-designed craft, we shall be in a position to try to improve conditions. Perhaps they are shy about asking for an engine that will streamline perfectly into their ships, while they still leave the entire landing-gear sticking outdoors, and are afraid we will point out to them that this is a crudity past which the birds evolved many thousands of years ago. There are many things we should tell the airplane designer loudly and often, and many that we wish they would similarly tell us. A few of these items are listed.

Engine Designers Need Airplane Data

We supply engine-installation drawings showing what the plane designer must consider when he sets out to attach our product to his, which indicate in a general way where we stop and where he begins. We would like a few sample airplane-diagrams showing, for example, the relation, size, and shape of the body that is to be attached behind our engine, and what constitutes good streamlining for the cowling in between, and how much of this should be our funeral. We also like to know if the plane constructor intends to tuck the oil tank against a bulkhead and arrange all the outlet louvers in the cowling ahead of the tank, so that an oil cooler will be needed, or if he will arrange for all the air entering the cowling to circulate round the oil tank. Numerous airplanes at the International Airplane Show, St. Louis, February, 1930, had their oil tanks carefully protected from any air-flow.

Will the airplane designer consider the engine mount to be a complete structure in itself and one that truly fits the engine; or must the engine, when bolted into place, rigidify and true up a welded assemblage of tubes?

For radial engines, does the airplane designer prefer a front or a rear exhaust-collector and, if the former, is his preference merely because he sometimes gets it with the engine and has not seen a standardized streamlined rear-exhaust system?

How far does the airplane designer want to go with National Advisory Committee for Aeronautics cowling, Townend rings, or other expedients for smoothing out the air-flow from the engine, and would he like these to incorporate the exhaust collector, and would he pay for such extras if we would supply them?

Will some airplane operators come forward and express their views on what percentage of forced landings are chargeable to (*a*) weather, (*b*) engine trouble, (*c*) fuel system, (*d*) installation, (*e*) plane structure, and (*f*) cockpit trouble? I hope someone who has had great experience will rearrange those items in the order he thinks they should have. What number, if any, of forced landings per 1000 hr. of operation would satisfy present safety requirements, and how much worse than the allowable minimum are the best present powerplants?

If an increase of say 0.5 lb. per hp. would effect a 3-to-1 reduction in the occurrence of forced landings, is it agreed that the increased safety would be worth the added weight? What changes should be made in the order of importance given to features such as reliability, durability, specific weight in pounds per horsepower, ease of maintenance, first cost and fuel consumption?

Performance Discussed

Weight and drag for a given horsepower are prime factors which determine airplane performance. The latest engine and cooling installations probably constitute less of the total parasite resistance of the plane than does the average landing-gear. Ultimately, both will be disposable inside the streamlined envelope of the ship. The N. A. C. A. cowling for radial engines, and the greatly reduced radiator sizes associated with high-temperature cooling, are steps in this direction so far as the engines are concerned. Corresponding improvements in landing-gear resistances seem to be limited to a few examples of "tin pants." Since there is no escape from lifting the wheels on an amphibian, satisfactory mechanisms have been put into production, but the gratis performance-increase from complete retraction is still in the future. In the meantime, however, we find that streamlined rocker-boxes on air-cooled engines are a good sales argument.

Results comparable to the reduced drag of our N. A. C. A. cowling are reported with the simplified British Townend ring, which is attractive on account of its small length, giving less interference with accessibility and avoiding doubt as to cooling interference. It has the marked advantage of not requiring a special design for each plane. If one must interview the airplane designer before one can design a cowling to ship to him, one is not going to ship any cowling in the present state of the art. Some allege that the openings afforded under the ring between the cylinders give marked advantages in visibility, as compared to the complete N. A. C. A. type. The best section and location for such a ring cowl is another aerodynamic problem on which the engine men would like to hear from the experts; because, if anything has to be hooked onto the engine, we prefer to attend to it ourselves.

The head resistance of an engine when considered by itself is meaningless and would, in general, exceed the combined drag of a well-cowled engine and fuselage. Frontal area for a given horsepower may be a crude criterion and, on this basis, the inverted V-type air-cooled engine is attractive, particularly for the most common type of installation, the single-engine tractor. Given the basic requirements that the propeller must clear the ground, and that the fuselage is in effect a streamline which must include a pilot with his sight-line above the engine, the best shape for the fuselage is a narrow body extending down from the propeller center, but not below the pilot's feet. The inverted V-type engine meets this condition nicely, affording excellent visibility and streamlining easily into the fuselage. It should be noted, however, that the necessary frontal area of the fuselage sets a limit to the advantage of further reduction in frontal area of the engine.

Reasons for Supercharging Stated

Improvements in power-weight ratio, either by lighter design or increased output, are necessarily gradual developments because of the extensive testing

necessary to assure that reliability will not be lessened. Supercharging is one of the most successful expedients for the following reasons:

(1) Sufficient experience has been had with the high-speed centrifugal type of supercharger, built into the engine, to demonstrate that—with the inclusion of a satisfactory shock-absorbing connection to suppress the otherwise enormous momentum effects—reliability is not in practice sacrificed.

(2) High-speed centrifugal superchargers improve the atomization and distribution, thus eliminating dependence on multiple carbureters and on inlet-manifold convolutions which are extremely critical as to distribution.

(3) Supercharging affords a wide range of increase in volumetric capacity, and therefore in horsepower, without compromising reliability by increased valve size or engine speed; and this is secured by the mere selection of the appropriate supercharger gear-ratio.

(4) With improved fuels becoming increasingly available, the safe degree of supercharging continually increases. The limit with any particular anti-detonation value in the fuel is set by the excellence of the cooling of the cylinder-head, and particularly at the exhaust-valve seat and stem and, in the larger bores, of the piston-head. Continuous improvements in these respects permit corresponding increase in power without increase in cost or weight, an extreme illustration being furnished by the winning engine in the last Schneider cup race, in which the initial rating was doubled, principally by supercharging. I am not advocating that degree of supercharging as a practical policy for standard-production engines today, but it shows how far we can go.

(5) The horsepower-weight ratio of an engine may be expressed by the number of cubic feet of mixture handled per pound of weight, on which basis the capacity of the centrifugal supercharger is enormously superior to that of the piston-displacement elements of the engine. If we consider them as air pumps and compare the piston type with the supercharger as to weight of the structure and quantity of air compressed, the advantage is enormously in favor of the small fan.

In some quarters a supercharger is still regarded as a doubtful complication, although, on many radial engines, it has permitted the substitution of a single carbureter for a triplex induction-system. This typical objection to so-called complication, on general principles, calls for comment. Life does continuously become more complicated, and the airplane itself is one of the latest additions, but we should be the last to condemn it on these grounds. One modern magneto has more parts than the entire engine which was fired by the original hot-bulb method, but we are forced to accept the modern development. A watch is just as great a mystery to me as an airplane engine may be to a watch builder, but we each benefit from the other's product in spite of its complexity.

The addition of parts giving increased performance or reliability is only to be criticized by those who have obtained the same results by simpler means, a condition which unfortunately seldom occurs. It is admitted that each added part is another item requiring careful design, testing, fabrication and inspection, but that is what the engineer and his associates are for. Failure in "simple" and in "complicated" mechanisms occur from the same cause; that is, from parts defective in design, material or fabrication and, from this viewpoint, two engines represent more complication than one, but no one would wish to limit the number of engines used on this account.

In any event, it is only relatively new complications that worry us. We are accustomed to 12 cylinders per engine for example, in which many of us prefer 48 valves to 24 on the ground of reliability, but we do not like to hear that the march of progress will force us to worry out satisfactory solutions for such problems as completely retractable landing-gears, variable-area or camber wings, variable-pitch propellers, vibration dampers and other contraptions, especially as we are now in the process of digesting reduction gears, inverted engines, and even the use of ethylene-glycol instead of water.

Reliability Defined in Safety Terms

From the viewpoint of safety, reliability means the elimination of such derangement, breakage, deterioration or wear of any part as might cause a sudden power-loss of more than say 30 to 50 per cent. It may be noted that, except for fuel and oil failures and "cockpit trouble," all engine failures are occasioned by the failure of some part or parts. All parts failures are chargeable to design, fabrication, assembly, maintenance, or material. On any widely used type, operating experience enables the engineer to correct the design progressively, while the other items come under inspection or quality control. "Uniformity control" is a still better term because, with any particular proved design, increased material strength is of no particular advantage and may be objectionable.

The requirement for reliability is the elimination of all specimens with properties outside of the specification tolerances. With modern methods of heat-treatment control, the outstanding problem is the detection of local defects such as hidden hair-cracks, inclusions, porosity and the like, which are the points where fatigue failures get their start. Due to the rapid load-variations to which almost every part is subject, breakages in aircraft engines are almost exclusively fatigue failures.

With adequate operating experience, relatively little difficulty exists in developing the design of each part so that the operating stress-range is safely within the fatigue limit of homogeneous material. The elimination of local defects, often of microscopic proportions, is a problem in metallurgical inspection which is in process of solution. The finished product of the steel maker is raw material to the engine builder, who imports most of his material failures with it because the defects are localized in minute fractions of the total bulk of the material.

Quality of Materials Considered

The "quality" of modern alloy-steels, in the sense of improved physical properties, is excellent in normal specimens, but the weight reduction which this should allow will not be realized successfully until this quality can be guaranteed by inspection to extend through every element of every piece. The electro-induction inspection-method has solved the problem for bar stock, and when an equivalent technique for forgings and castings has been developed and applied, accidents from material failures should cease to bother us.

With regard to castings, one British designer takes the foregoing situation so seriously as to carve elaborate cylinder-heads and crankcases from forged duralumin masses. In this Country, we feel that prop-

erly designed castings eliminate the necessity for such great additional expense, although one manufacturer has reduced his crankcase halves to such extremely simple form as to be producible by forging, with only the normal allowance for machining of fitted surfaces. However, even this excellent job seems to realize extra strength rather than a marked reduction in weight.

When the quality of the material is uniformly poor, it is easily detected and the limiting stress for fatigue endurance is indicated by the low tensile-strength. Any sudden discontinuity, whether from a material or design defect, will generate severe local deflections which may be compensated by the ductility of the material under a single slowly applied test-load. No material, however, will withstand continuous repetition of such local deflections which will produce a crack gradually growing from the defect until the familiar fatigue fracture is complete. In uniformly good material, such cracks can occur only when the piece is stressed in service beyond the well established fatigue-limit of the material; then, the crack will start from the sharpest corner, or from the deepest scratch in the region of excessive stress. If the general section of the part is definitely overstressed, removal of the scratch or sharp corner will not prevent ultimate fatigue-failure, but will merely render the location of the crack erratic. It is astonishing how easily a sharp-corner effect may be overlooked in the design of a complicated piece, until a fatigue failure calls attention to it, and such effects are one of the worst oversights of which the designer can be guilty.

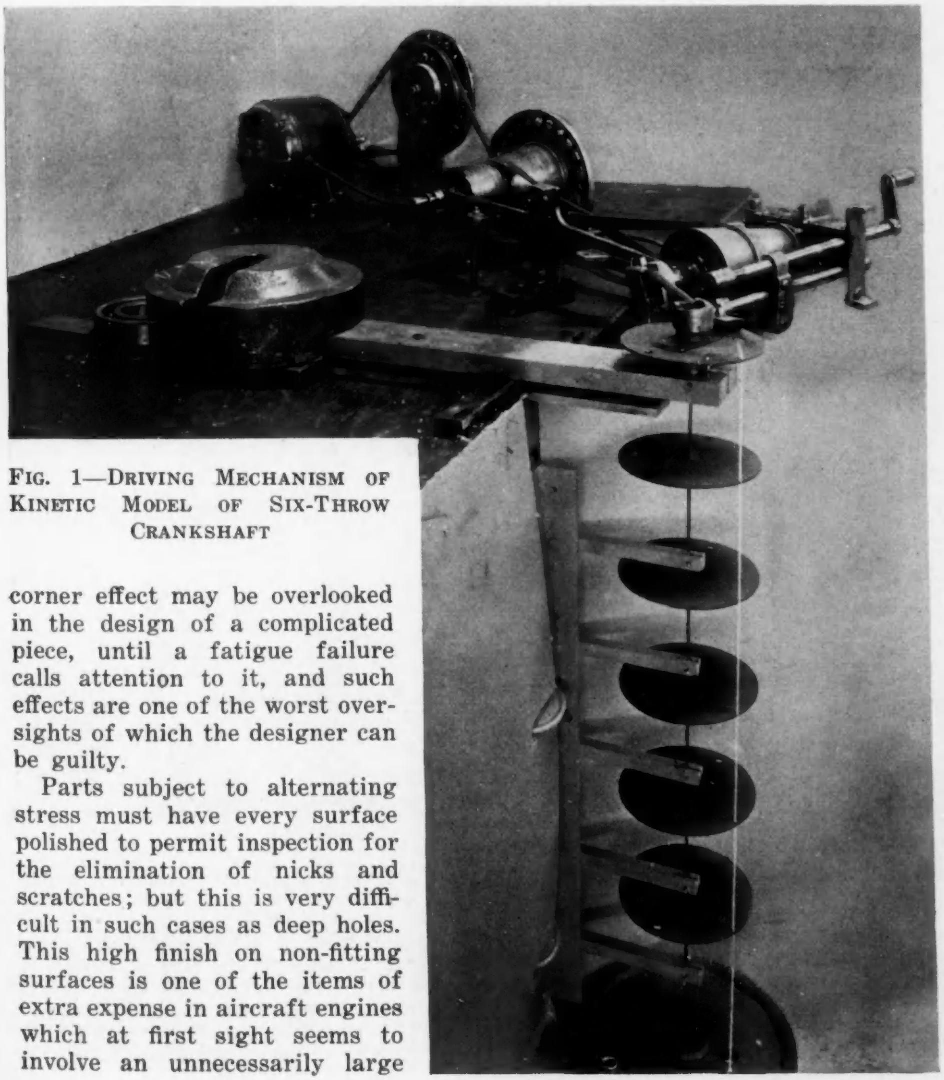

FIG. 1—DRIVING MECHANISM OF KINETIC MODEL OF SIX-THROW CRANKSHAFT

Parts subject to alternating stress must have every surface polished to permit inspection for the elimination of nicks and scratches; but this is very difficult in such cases as deep holes. This high finish on non-fitting surfaces is one of the items of extra expense in aircraft engines which at first sight seems to involve an unnecessarily large percentage of rejections. Until cheaper and surer methods than visual inspection are perfected, the routine test, tear-down, re-inspection of all parts, re-assembly and check test on each engine, are additional items of expense incurred in the effort to realize maximum reliability. These short tests, however, do not afford much warning of impending fatigue-failures for which long endurance-testing is essential.

The minute examination of parts from service or endurance engines for the beginning of fatigue cracks is of great value, and we anxiously await the development of some more rapid and sure method than etching for detecting these incipient failures. The development of fatigue cracks is a lengthy process, as is exemplified by the former railroad practice of periodically inspecting car-wheels by merely observing the "ring" when struck with a hammer. If a corresponding test ever should be developed that is applicable to the parts in an assembled engine, our fears from material defects would be largely eliminated. But the routine tear-down inspection does serve to perfect the assembly specifications and inspection as to standardized adjustments, tolerances on fits, and the like.

Wear is less of a safety problem because the symptoms usually can be detected by routine service-inspection, while complete seizure or bearing failure is not endangered by obscure local defects in the material, except possibly on ball-bearings. One remaining cause of sporadic failures resides in parts which are on the ragged edge of being overstressed, and this is discovered only when some abnormal condition increases the local load. Here, again, experience is the cure.

Effect of Thermal Conditions Analyzed

Thermal conditions, as in exhaust valves and piping—particularly in air-cooled cylinder-heads—deteriorate the properties of the material and cause erratic failures. There is room for great improvement in air-cooled engines in this respect, and careful researches in exhaust-port and fin design indicate that these improvements will soon be realized in practice. The mean cylinder-head temperature on present designs is already at a safe figure, the elimination of local hot-spots being the outstanding requirement as to both reliability and performance. Guiding the air-flow to the critical points is an obvious expedient and a 100-deg. fahr. drop in temperature at the rear spark-plug boss recently has been attained with relatively crude deflectors. By adding another device, that drop in temperature has since become 200 deg. fahr.

Synchronized Vibrations Dangerous

Consistent failures are easy to eliminate from all parts of the engine, and the only common cause of erratic failure not already mentioned resides in parts subject to resonant vibrations which result whenever the inherent natural frequency of the part, as predetermined by its mass and elasticity, synchronizes with the periodicity of the load to which it is subject. The glass bowls which may be shattered by the striking of a synchronous piano-note are a scientific curiosity illustrating this form of failure. Crankshafts, valve springs, pipe lines and supercharger drives are examples painfully familiar to the powerplant engineer.

The period of vibration in a part is determined when it is designed. The period at which it will vibrate is not influenced when it is installed in the engine, no matter how it is operated. The only question is: Will the operating impulses synchronize with that natural period of the part? If they do, the engine will go to pieces.

A crankshaft may be constructed of material having an ascertained safe fatigue-endurance stress of 40,000 lb. per sq. in. while it is impossible to predict any operating stress higher than 10,000 lb. per sq. in.; yet, one failure may occur in a few hundred shafts. The failure depends entirely on how long the engine happens to be operated at a speed synchronous with the natural torsional period of the shaft system and has no connection with calculable stresses from inertia and explosion loads. In the Graf Zeppelin, for example, a coupling change brought the natural period of the shaft system into synchronism with the cruising-speed impulses; three crankshafts broke on a single flight, but the fatigue cracks in the fourth had not quite reached the point of complete failure. That is about the most consistent example of fatigue failures in crankshafts of which I know. "Beefing up" the crankshaft is the

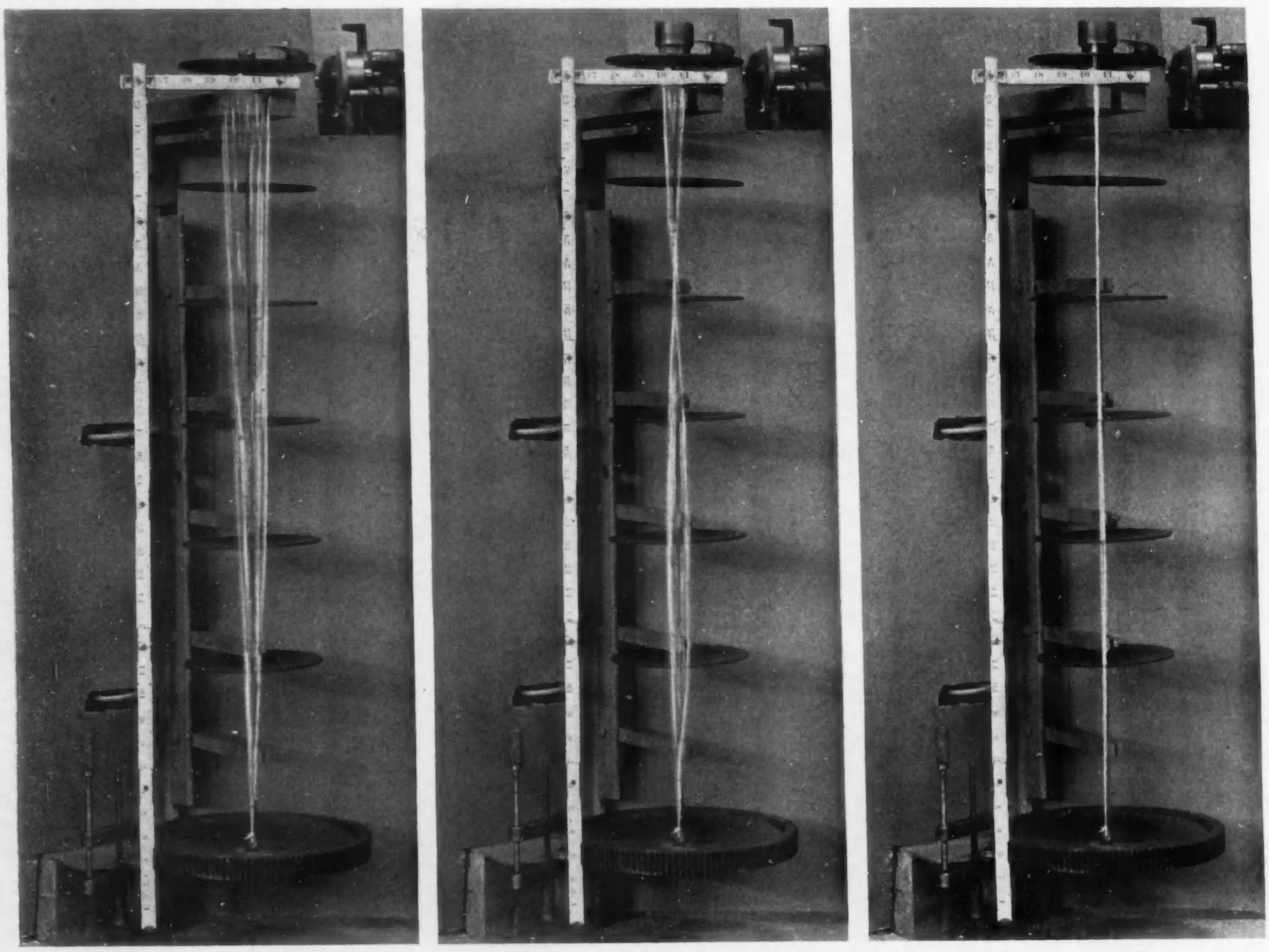

FIG. 2—KINETIC MODEL OF SIX-THROW CRANKSHAFT

The Small Discs Represent the Six Crank-Throws and Associated Parts. The Large Disc Represents the Propeller. The System Is Caused To Vibrate by Light Frictional Contact of the Top Disc with an Oscillating Torque-Arm Driven Through a Variable-Speed Gear by an Electric Motor. The Vertical White Band Shown Is Produced by the Movement of a Normally Vertical String Attached to the Edge of Each Disc and Indicates the Amplitude and Nature of the Oscillation. The Points at Which the String Remains Stationary Show the Location of the Nodes

The Stiffness Ratio of the Liberty Model Is $830/0.01017=81{,}500$ and the Mass Ratio Is $11.34/0.186=60.9$. Therefore, the Frequency Ratio Is $\sqrt{(81{,}500/60.9)}=36.5$. The Rate in the Left View Was 2.35, That in the Central View Was 7.08, and That at the Right Was 6.70 Cycles per Sec. These Tests Were, Respectively, To Develop Primary Torsional Vibration (Left), Secondary Torsional Vibration (Middle) and Behavior at the Non-Resonant Torsional Impulses (Right). Preponderance of Vibration Effect Is Obvious, Since Stresses Are Proportional to Deflection

expedient usually followed. When this is successful it is not because the extra strength will enable the shaft to withstand torsional vibrations, but because the stiffening of the shaft has brought its period above the operating speed-range of the engine.

Means of Measuring Torsional Vibration

Fig. 1 shows a kinetic model of a six-throw crankshaft, by means of which torsional vibration can be illustrated. Fig. 2 indicates the set-ups used and states the needed data. Fig. 3 shows the equipment used for demonstrating crankshaft torsional deflections.

It should be noted that under torsional vibration the reverse torsion on the shaft is as great as the forward torsion and both will exceed many times the normal peak-torque of the engine. Some years ago I constructed a simple torsion-meter with which I measured torsional vibrations in excess of $\pm$ 5 deg., three times per revolution, in a Liberty crankshaft at its critical speed. The free end of that shaft is going ahead and lagging 5 deg. behind the propeller at the worst point in the torsional period three times per revolution; so, any gearing installed at the end of that shaft has hard service. The normal over-all deflection in a Liberty crankshaft due to the peak explosion and inertia torque is less than 1 deg. and, since the stresses are proportional to the deflections, the preponderance of vibration effect is obvious from the foregoing figures.

These defects are emphasized, and become audible, when a transmission is appended to the engine; hence, automobile manufacturers have been forced to incorporate some form of damper, usually the simple friction-driven-flywheel mass at the free end of the crankshaft which was originated by Lanchester, who foresaw its necessity 20 years ago.

The Air Services now realize that no crankshaft, regardless of its size, will remain in one piece for more than a few hours at synchronous operating-speed. They probably will require a demonstration by means of a torsion meter that the natural frequency of the shaft is outside the operating range, or else that it be suppressed for all speeds by some form of vibration damper. I am strongly in favor of the adoption of the second expedient, but it remains to be seen what is going to be approved by the industry.

Lightening the shaft to bring its period below idling speed is not practicable with directly mounted propellers, but it is possible with a long transmission shaft as has been proved by motorboat installations in which the power from an engine having a 3-in. crankshaft—which has been known to fail with a directly connected driven mass—is successfully transmitted by a propeller-shaft as small as 1¼ in. in diameter. In other words, crankshaft failure can be avoided by the inclusion of a sufficient length of shaft of more than 10 times inferior strength or by the use of any equivalent torsional spring-drive. Such a spring means is incorporated in the geared Curtiss Conqueror engines. I am not indicating that the yield of the spring in this engine is as much as that in motorboat propeller-shafts, but it is tending in that direction.

The foregoing point is also nicely illustrated in the supercharger drive on the inverted-V air-cooled engine developed by our company. The drive is by a long shaft from the constant-speed end of the crankshaft, the propeller end, to the rear of the engine. No trouble whatever has been experienced with impellers, gears or their shafts and bearings, at impeller speeds as high as about 25,000 r.p.m. This is in marked distinction to the case in which it is attempted to gear up the impeller rigidly from the free end of the crankshaft, which may be accelerated several degrees several times per revolution.

By examination of the crankshaft driving-gear teeth from such an installation, it is easy to see with what number of cylinders the engine was equipped. The "footprints" of each explosion impulse are clearly spaced around the circumference of the gear, and the midway points show equally high loads on the so-called non-driving side of the teeth. In other words, the erratic rotation of the free end of the crankshaft entirely overcomes the steady load of perhaps 20 hp. required to drive the impeller, which is literally kicked around by a succession of blows on a few of the teeth. A simple friction clutch cures this condition. I prefer a form which generates somewhat more than the necessary horsepower torque in the forward driving direction only.

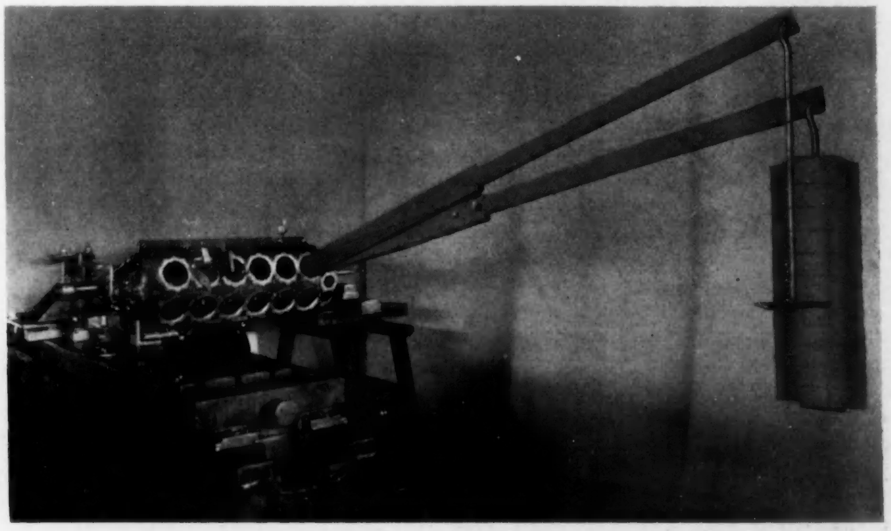

FIG. 3—TORSIONAL TEST OF LIBERTY-ENGINE CRANKSHAFT IN A CRANKCASE
The Double Exposure Was Made To Show the Torsional Deflection of the Shaft Under Load

Valve Springs and Valves Considered

Helical valve-springs are particularly susceptible to synchronous vibration-periods of the middle coils which persist with the valve both open and closed. Many have a period around 10 times per valve cycle, which means that a range of 5 per cent in engine speed must include a synchronous period. That is the difficulty when these periods become very high compared to the engine speed. It is almost hopeless to avoid them; they must be damped.

The laminated flat springs, in bending, such as have been used on several types of Aeromarine engines, include in their advantages a completely "dead-beat" ac-

tion; accordingly, no failures are known to have occurred with such springs. Some simple means of obtaining corresponding damping of helical springs would end the troubles with these units.

Similarly, a reliable pipe-coupling which would destroy the elasticity in the cantilever elements of fuel lines would eliminate a prime source of fire hazard. As with the crankshaft-vibration damper, it is likely that the solution of these problems was experimentally achieved long ago. If so, we have two more "complications" which should be incorporated in our product.

Valve failures afford another instance with which the obviously calculable loads have little connection. Impact or pick-up velocities, in comparison with the mass and the safe elastic yield of the parts, determine the stresses. It will be seen that, when high seating-velocities occur, the valve-stem mass tends to continue traveling; in fact, it actually will do so after prolonged repetition of excessive impact. With overhead camshafts or fully compensated push-rod gears, moderate valve-clearances can be used and the seating velocities kept down to a moderate figure, say to $1\frac{1}{2}$ ft. per sec. Non-compensated gears on radial engines may involve valve-clearance variations from hot to cold of as much as 0.050 in., which makes it difficult to secure low seating-velocities with good timing-angles.

Effects of Increased Clearance

Corresponding increase in clearance due to wear is equally hard on the valves. Little trouble is experienced in this direction with oiled valve-gears such as are obtainable on in-line engines, but the lubrication of parts in rocker boxes presents a serious problem, including the maintenance of moderate temperatures and grease tightness in the box. This has not yet been achieved except by abandoning valve-clearance compensation in favor of the integral rocker-box construction, which can be designed to improve the cooling greatly. Current developments by our company include a design predicated upon the combination of these features, and on the consideration that, with excessive valve-clearances, ultimate failure of an otherwise excellent valve gear is inevitable.

The comment has been made to me that this statement boosts the in-line engines, but I do not think so. The valve gears on radial engines constitute a more difficult problem than do the valve gears on in-line engines, but we think the radial valve-gears can be brought up to equal reliability and that this is now being done. This is not a discussion as to the merits of different kinds of engines.

The principal stress on the valve being that due to the mass of its stem, no theoretical gain in strength can be had by increase in diameter, while increase in length —which may reduce the temperatures at the valve tip— is in other respects distinctly disadvantageous. These theoretical deductions are well borne out by the practical results of variations in the proportions discussed. The combination of good guide-cooling with light valve-stem weight is the solution. In air-cooled engines the latter is improved at the expense of the former by the use of dual exhaust-valves. Some persons will not agree with that statement. The advantage of using dual exhaust-valves is that each valve is of light weight. The disadvantage is that it is difficult to install enough cooling fins around the exhaust ports, because these must be so close together. One successful British engine incorporates this feature, and even seems to have good durability with such lubrication as can be had with an exposed valve-gear.

Must Base Designs on Operating Deflections

Some of the foregoing remarks emphasize the point that many design problems are involved in considerations of operating deflections, and the engineer who is technically successful must keep in mind continually that every inch of steel in the parts undergoes a deflection of 0.001 in. for each 30,000 lb. per sq. in. of stress increment. This is a moderate alternating stress and it is useful to form the habit of visualizing each element of our so-called rigid parts as continuously vibrating through about 0.002 in. with respect to points 1 in. distant, and proportionately farther in the case of greater separation. From this viewpoint, every structure in the engine has all the properties of a spring, and we often get into trouble by failing to realize to what substantial distortion the "spring" sometimes amounts.

For example, the relative excursions measured tangentially between the end crankpins in the Liberty-engine crankshaft at its torsional elastic-limit actually exceed the operating deflections of the valve springs, while the energy elastically stored in the deflected shaft amounts to more than 1000 ft-lb., which is more than the total energy from one explosion. Incidentally, it should be noted that the amount of spring is virtually independent of the quality of the steel, heat-treatment making virtually no change in the modulus of elasticity. No specially "stiff" piece of steel has yet been produced, increased tensile properties merely bringing greater deflections to be within the elastic limit. This simple fact enables operating-stress measurements to be obtained from parts of a complete structure with an instrument which merely records the elongation between any two points as in the ordinary turbine torque-meter which records the horsepower load by measuring the twist on a few feet of a marine propeller-shaft. This instrument has a tube that goes around, say, a 22-in. marine propeller-shaft and measures the torque between two points located about 2 ft. apart.

Importance of Deflections Instanced

Lubricated bearings afford an example of the importance of deflections because they depend on the maintenance of a definite oil-film-thickness variation between the loaded and unloaded sides. If the bearing should actually "fit" the journal over a substantial arc, the automatic entrainment of oil, by which the rotating shaft maintains the oil-film pressure equal to the intensity of load on the bearing, is destroyed. Metallic contact then occurs which will completely destroy the high-duty main-bearings in an aircraft engine in a few seconds. The required contour relationship between the journal and the bearing must be expressed in fractions of 0.001 in. and can be completely destroyed by corresponding deflections in the bearing shell. For steadily loaded bearings, relatively soft lead-and-tin linings may squeeze out to compensate for the deflections in the housing; but, under variable loading conditions, repeated deflections promptly crack such linings.

Unfortunately, deflections occurring in ordinary bearings usually tend to conform the radius of the shell to that of the journal over the loaded zone, thus destroying the essential clearance. One can consider almost any bearing combination, visualize it under the load it carries, and then try to visualize the deflection

in the surrounding materials. It will be found that almost always those deflections are tending to warp the bearing shell and to deflect it enough to make it really fit the journal; that is, to conform it to the same arc. When that happens the bearing is gone, whether it has force-feed or any other kind of lubrication.

Bearings constructed with separate facing-segments that automatically compensate for deflections in the housing while maintaining the necessary difference between the pressures and clearances at the "toe" and the "heel" have been developed by Mitchell and by Kingsbury from theoretical investigations of the theory of film lubrication. These bearings have been successful in many commercial applications, including thrust bearings of which the oil plain-collar type were incapable of withstanding any substantial specific pressure. This is another development, however, which has not yet percolated into the airplane industry; which, accordingly, still depends on mere massiveness to make its bearings work.

Influence of Piston Operation

The automatic film-forming properties inherent in journal bearings are unfortuntaely absent in the case of the piston, where the tipping of short and loosely fitting skirts tends to throw the preponderance of pressure on the leading rather than at the trailing edge; hence, the piston has the most uncertain bearing in the engine. In steam-engine practice no corresponding trouble occurred because the piston, guide slippers, and fire box are segregated. The concentration of these functions into the cylinder of an internal-combustion engine adds a difficult thermal problem which is partly met by thick piston-heads of material having high conductivity for dissipating heat to the cylinder-walls. An appreciable amount of heat is abstracted from the piston-head by the oil spray from the crankcase, and this benefit can be extended as the control of oil past the piston-rings is improved, probably to the desirable extent of making oil radiators very effective on the pistons.

Constant-clearance pistons, as already standardized in automobile practice, will help to solve both piston-bearing and oil-pumping problems. The piston-rings themselves are a further illustration of the action of oil-films, as they tend to ride over the oil when the leading edge has the lower intensity of pressure, and to scrape oil when the pressure on that edge predominates. Here also the difficulty lies in maintaining the desired minute dimensional differences in spite of warpage, wear and piston tilt. The difference between a piston-ring that pumps while going up and one that pumps while going down is probably a matter of 0.0001 in. and is difficult to control.

Diesel Aircraft-Engine Development

The solid-injection type of compression-ignition engine has attracted great interest since the Packard Diesel-engine-powered airplane flew from Detroit to Langley Field at a very low fuel-consumption cost. Intensive scientific investigation of fuel sprays at the laboratory of the National Advisory Committee for Aeronautics has reached the point where fuel charges of 85 per cent of the full-power quantity have been completely burned at an indicated mean effective pressure of 133 lb. per sq. in.—or about 110 lb. per sq. in. brake mean effective pressure—up to a speed of 1500 r.p.m. and engendering maximum pressures of 760 lb. per sq. in.; but these results do not yet compare favorably with those of the conventional spark-ignition or constant-volume cycle. The desirable constant-pressure combustion-characteristics obtained by Diesel with large injection-air compressors have not yet been realized in high-pressure solid-injection engines.

Only about 0.0025 sec. is available in which to atomize the fuel and to get the mixture fired and burned before any of the fuel particles touch the cylinder-wall. This time-lag corresponds to 25 deg. of crank travel at 1500 r.p.m. In other words this is, roughly, the time available for atomization of each fuel injection, which should permeate the entire combustion-chamber but burn before touching the cold walls. A time lag of the order indicated is also observed before the fuel reaches the injection orifices under an injection pressure of 4000 lb. per sq. in. The fuel pumps and injection nozzles, which must function in these minute dimensions displace the present carbureter and ignition systems, and seem to entail considerable development to be equally reliable. The required fits on some of the valves and plungers are expressed in millionths of an inch.

I venture the prediction that an easier method of burning crude oil will be found in carbureting the fuel with the air after compression, in a device similar in theory to that of the present carbureters but adapted to the greatly increased pressures, temperatures and densities which seem to me to be precisely what are needed to handle the less-volatile fuel.

Other Difficult Problems Cited

Noise reduction is another impending problem of minor importance but major difficulty. The development of a muffler that is effective without producing substantial back-pressure and is capable of withstanding the exhaust temperatures will be merely a preliminary step. Silencing the terrific noise which will then be evident from the engine valve-gear, piston-slap, supercharger and reduction gears is a larger problem than is the one that has required several years of concentrated effort in quieting automobiles. The propeller alone produces a noise which, as the tip speed approaches the velocity of sound, completely dominates the open exhaust and the engine noises. Remoteness of the engine from the cabin, and insulation of its vibration to prevent sound transmission through the structure, will be of great assistance. If the quietest automobile engine were placed inside the closed body, the noise at high speed would be unbearable. When streamlined airplanes are built, the resulting removal of the present great air-disturbance noises will unmask further engine noises.

Comments on Reduction Gears

Reduction gears have been developed to a point of satisfactory durability without yet being very generally used by the airplane builders who originally requested them. The ones principally produced are of the bevel-gear planetary-type initially developed by Farman in Europe and, in this Country, by me. The Wright Aeronautical Corp. has standard gears of the former design available in two ratios for all its current engine-models, and the Pratt & Whitney Co. uses the latter design.

Because of the high tip-velocity, which is often 600 m.p.h., the propeller constitutes a very large flywheel. Illustrating the result by analagous automobile-practice, the propeller-shaft therein is able to transmit three or four times the engine torque as smoothed out by the flywheel. Corresponding capacity in the con-

nection between the crankshaft and the flywheel, however, would be wholly inadequate.

Airplane-engine reduction-gears comprise such a connection through which the crankshaft "bounces back" from the propeller mass when the shaft "unwinds" after each explosion impulse, and this is about 150 times per sec. in a nine-cylinder engine. Elastic energy in the shaft is thus transformed into flywheel energy in the propeller at the frequency indicated, and vice versa. The energy fluctuation to which the gear is thus subjected may exceed the total explosion energy of one cylinder when torsional resonance is encountered. Naturally, the driving sides of the teeth do not give evidence of much greater punishment than do those which we humorously term the "non-driving" sides. The most perfect gears will be noisy under these conditions. All of these destructive effects would be eliminated by a suitable damper whereby the gear would need only to transmit the torque due to the power developed by the engine.

Author Seeks Constructive Criticism

In anticipation of comment that some of the foregoing remarks on engine design are too theoretical to merit practical action, it may be mentioned that most of them have some bearing on current or projected developments at the Wright Aeronautical Corp., and it is hoped that those to whom they cover familiar ground in an elementary way will come forward with their criticisms. It also is possible that even those who seek safety by strict adherence to proved details will find herein something of interest, in view of the fact that seemingly slight changes in environment or conditions may have far-reaching results upon the performance of a part, explicable only by application of the appropriate theory.

For instance, the consistent failure of a previously satisfactory part in the first installation in which synchronous phenomena are encountered will puzzle anyone who regards the theory of resonant vibrations as something invented by visionaries for their own amusement. The valid criticism of a theory does not consist in pointing out that it is theoretical, but in producing evidence to show where it is wrong. On this point a quotation from the history of the French Revolution is appropriate: "Jean Jaques Rosseau wrote a book called the Social Contract. It was a theory, and nothing but a theory. The French nobles laughed at the theory, and their skins went to bind the second edition of the book."

THE DISCUSSION

CHAIRMAN B. G. LEIGHTON[2]:—The aircraft industry is very new. We have a production capacity about five times larger than our market. One way to increase the market is to reduce costs, and the best way is to find out how to eliminate the things which, considered as to final outcome, constitute wasted effort.

The purpose of the Council in establishing the Aircraft-Engine Activity as a separate activity is, in my opinion, to gather together those who are interested in development and afford them an opportunity to exchange ideas. One of the Past Presidents of the Society pointed out that in practically all new industries at the outset, each engineering company thinks that it has a great secret and the tendency is, therefore, for various individual engineering companies to keep their affairs secret, fearing that some other company will get ahead of them. To quote the Past President's statement, "That theory is all wrong. I have one idea and you have one idea. If we each keep it secret, we each have but one idea; but if you tell me your idea and I tell you mine, I have two ideas and you have two ideas and we both derive benefit."

At the St. Louis Aircraft Show, we have about twice as many models of airplanes as the industry needs. We are playing around with so large an excess because we do not know what the aircraft operating-companies are likely to want or what the public is likely to want. There should be some means of arriving at lines of effort upon which we can all agree, so that as a complete industry we can stop wasting money and effort on non-essentials and sell a sufficient number of airplanes so that we can all get down to a profitable basis of operations. I believe that we can do this only by free interchange of ideas and open discussion. That course will require that we contribute information of value, but I believe that we shall all benefit.

ROLAND CHILTON:—Regarding the valves of an engine, the heat from the head may be conducted to the seat, which is relatively cool; or conducted up the stem to the guide, *if* the guide is relatively cool, although this is seldom the case. The valve stem of large diameter has a large cross-section and the temperature gradient up the stem is reduced. The disadvantage is that the stem is heavy. The compromise, which is the general practice, is to use salt which melts, splashes up and down the stem, and transmits the heat from the bottom of the valve to the top and to the valve-stem guide if that is well cooled.

In general, air-cooled engines are not as cool as we like to have them. The heat-dissipating capacity of the valve-stem guide may be very disappointing. In addition, the guide is a bearing and there is a limit to the amount of heat which can be dissipated through a bearing, especially if there is no visible means of lubrication. If the tip of the valve runs hot, there is the question of valve-head weight. The ideal is to use the shortest possible valve stem and have the guide as close to the valve head as possible. The point I want to emphasize is that a long valve-stem is definitely bad practice. The valve stem should be as short as possible to obtain satisfactory durability of the guides. The amount of wear on the guides depends primarily upon how much side pressure exists, and the amount of side pressure depends upon how the gear is designed. With a rocker-arm gear, where excessive side-pressure is exerted, it may be necessary to install a very long valve-stem but it is bad practice in every other respect, except that possibly it keeps the valve-tip temperatures from becoming too high.

J. H. GEISSE[3]:—Change in timing with change in valve clearance is not generally understood and we have an exhibit at the St. Louis Aircraft Show which shows

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[3] M.S.A.E.—Vice-president in charge of engineering, Comet Engine Corp., Madison, Wis.

this effect. It is remarkable how difficult it is to explain to those who are actually operating the engines how to account for it.

The change in timing with change in valve clearance cannot be avoided except by compensated gears, because the cam must be designed to give a certain closing and opening velocity for the valves throughout the entire range of clearance that will be used in the engine from hot to cold. If it were possible to design the cam for a satisfactory closing and opening velocity at only one clearance, the cam could be designed so that there would be very little change in timing; but that cannot be done without danger of pounding the valves into the seats.

Mr. Chilton mentioned mufflers. We have a muffler that is satisfactory in that it is an entirely open tube that muffles very well and produces no back pressure. We tested that on the manifold when a propeller was installed on the engine, and we could not detect the difference in noise with the muffler on or off. The propeller noise is so great that it is practically useless to put a muffler on the manifold, although I know that it has been very satisfactory on other engines.

The design of the exhaust ring as a muffler is interesting. On the first exhaust-ring that we built, the exhaust was run tangentially on one side in one direction and in a different direction on the other side. That was entirely satisfactory from the performance viewpoint and, to the pilot, the engine was very quiet; but, to observers on the ground, the engine sounded more like a 500 than a 165-hp. size. We changed the pipes so that they ran in the same direction all the way around; then, when flying over the field, the engine noise could not be heard although one could hear the propeller.

P. B. TAYLOR[4]:—Some do not realize what is available in the way of additional power or reduced weight in an aircraft engine nor do they realize its relation to fuel consumption. With a given fuel, it is possible to increase the output of the engine if one is willing to sacrifice fuel consumption; and that is one reason why it is so important for the airplane manufacturers to tell the engine manufacturers whether they want power or low fuel-consumption, or both power *and* low fuel-consumption.

The use of better grades of gasolines, with which both additional power and low fuel-consumption are available, makes it possible to proportion these two characteristics in any ratio, within certain limits. One thing for which we are working is the adoption of better gasoline—if the public so desires—so that more power and less fuel consumption are both accomplished. The engine manufacturers and the airplane manufacturers together must interpret the opinion of the public and decide in just what direction engine development should be directed.

G. W. LEWIS[5]:—I wish to congratulate Mr. Chilton on his excellent presentation of the most important factors affecting the reliability of aircraft-engine operation. With reference to the cowling of air-cooled engines, it was noticed at the 1930 St. Louis Aircraft Show that, in most cases, the manufacturers of in-line type air-cooled engines have furnished the cowling. The radial air-cooled engine has many inherent advantages. I am sure that, as a type, it will be used for a long time and that the manufacturers of radial air-cooled engines will soon furnish cowling as a part of the engine installation.

The National Advisory Committee for Aeronautics has just completed an investigation covering a series of tests on cowlings for radial air-cooled engines, varying in width from 6 in. up to a full cowling. The report is not considered final, as the factors controlling the design of the ring cowling are different from those of the full or the N.A.C.A. type of cowling. In the ring cowling, it is very necessary to consider the direction of the air-flow so that the ring can be mounted so as to give the minimum drag with a positive downwash.

With cowlings provided by the engine manufacturers as part of the engine equipment, it will be possible for the airplane designer to design the fuselage properly, not only from the viewpoint of providing a streamline flow with the minimum of resistance, but also of taking care of the important factor of visibility.

Few Failures Chargeable to Engine Structure

The aircraft engine is still an important contributing factor in aircraft accidents. From the analysis of aircraft accidents made by the Aeronautics Branch of the Department of Commerce, the powerplant is responsible for approximately 15 per cent of the aircraft accidents. The principal contributing factors seem to be the fuel and ignition systems, which contribute respectively 4 and 3 per cent of the 15 per cent cited. However, it is encouraging to note that, from the analysis of failures of powerplants, less than 2 per cent is contributed by the failure of the engine structure.

The relatively few failures charged to the engine structure, resulting in aircraft accidents, is most certainly a credit to our aircraft-engine designers and manufacturers and is a measure, in a way, of the reliability of the engine structure proper. I am sure that, if the same care were given to the design and construction of the powerplant installation that is given the engine proper, the reliability of the powerplant would be increased and the number of accidents charged to powerplant failures would be relatively few.

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[5] S.M.S.A.E.—Director of aeronautical research, National Advisory Committee for Aeronautics, City of Washington.

Stringent Requirements *for* Aircraft-Engine Parts

St. Louis Aeronautic Meeting Paper

By W. F. WISE[1]

MEETING the highest requirements of the automobile industry for highly developed materials, accurately machined and properly heat-treated, proved to be but a preparatory school for meeting requirements of the aircraft industry. The modern automobile would not be possible without the special alloys that have been developed, together with the heat-treatment required to bring out their desirable physical characteristics; but no branch of the automobile industry, unless possibly the building of special racing engines, ever approached such demands upon materials of construction as the aircraft industry makes.

After reviewing the requirements which make the manufacture of aircraft-engine parts more costly than that of automobile-engine parts, the author gives particulars as to the care required in manufacture to guard against engine failure.

Inspection is ascribed a place of chief importance, including checking of material to specifications, and pickling and sand blasting as a preliminary to inspecting for surface imperfections. Milling threads and grinding all surfaces, even clearance holes, are recommended to guard against tool marks that might localize strains. Methods are described for checking heat-treatment.

In the discussion, the author gives reasons for preferring metallurgical analyses from an outside laboratory that is not prejudiced.

Although the reasons for this difference are almost too obvious to mention, some still have the feeling that an engine part is an engine part, and the mere fact that the part is intended for aircraft use should not cause an increase in manufacturing cost which frequently is between 50 and 250 per cent. Two reasons for this difference are that the permissible weight per horsepower of an aircraft engine is from one-fifth to one-tenth that allowable for an automobile engine; and that the automobile engine works at approximately 20 per cent of full power for 80 per cent of its running time, under conditions which make it virtually impossible for the maximum power output to be required for more than 5 to 10 min. consecutively. The aircraft engine, except while warming up on the ground, rarely delivers less than 75 per cent of its maximum power, and an engine which is not capable of 50 hr. at maximum power in consecutive 5-hr. periods is not considered worthy of the approval of the Department of Commerce.

If long periods of operation at high power output were not required, the mere lightening of the automobile engine to the standards required of the aircraft engine, without the use of higher rotational speeds than are permitted in aircraft, would introduce difficulties which would result in headaches for the automobile-engine designer and heartaches for the factory management that was required to meet the increased cost.

The removing of material required for lightening would introduce many expensive fixtures and operations not otherwise required; and the increase in unit stresses, even under automobile operating conditions, would require more highly developed materials and manufacturing technique or would result in a great increase in the number of fatigue failures.

When this lightening in weight is coupled with the ability to withstand long sustained runs at maximum power, the tendency to fatigue failures is still further heightened; so that the difficulties, and therefore the effort to overcome them, must be increased virtually as the square of the power output per unit of weight.

Manufacturing Provisions Against Engine Failure

Add, to the foregoing, considerations of the moral responsibility, because the failure of an aircraft engine involves the potential loss of from one to a score of lives, and of the fact that, on a purely commercial basis, the engine builder who has the lowest percentage of engine failures will meet with the broadest success, and it is easy to see the difference between automobile and aircraft-engine parts and to justify a considerable difference in unit cost.

Some of the more important precautions against expensive and disastrous engine failures may be outlined as:

(1) Selection of suitable material for a given part
(2) Correctness of the material to the analysis specified
(3) Freedom from mechanical defects such as roll seams, pipes, slag inclusions and forging defects
(4) Correct heat-treatment
(5) Accurate machining, grinding and production operations
(6) Thorough inspection

Mentioning inspection last on the list is not meant to infer that one inspection of the finished part is sufficient. The old adage, "Everlasting vigilance is the price of safety," may have been partially appreciated but was never fully realized until men began to fly.

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Rigid inspection and testing of each piece of raw material, continuous inspection of the manner of performing each operation, the inspection of the parts themselves after each operation and particularly every known inspection and test which will reveal any defects in heat-treatment must be performed with a thoroughness that would seem wastefully unnecessary in any other than the aircraft industry.

Because of the extreme importance of inspection and testing in connection with aircraft work and the fact that the ratio of inspection cost to total production cost is higher in this industry than in any other that can be called to mind, the functioning of the inspection department with reference to the various departments of purchasing, material preparation, machining, heat-treatment, grinding and final inspection will be taken up in detail.

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Material specifications for the various parts are transmitted to my company, after they are finally approved by the engineering department of the engine builder, with the order for the parts. In some cases, engine builders prefer to furnish the bar stock or forgings to the parts specialist. Whether or not this is done, the parts specialist should be provided with detailed specifications of the material to be used.

If we, as parts specialists, are to furnish the material, the engineering specifications for the material are forwarded to the mill with the order. The specifications will include the requirement that all material on that order shall come from one heat; shall be free from dirt and slag inclusions; shall meet certain requirements of machinability; and shall develop certain tensile strength, hardness, impact resistance and other physical properties when heat-treated. A specification for extended aluminum bronze is given herewith, as an example.

Checking Material to Specifications

Upon receipt of the material, it is checked as to analysis and machinability. Samples must be prepared for tensile, fatigue and impact tests, and suitable determination of inclusions must be made. This determination falls under the heading of material preparation and may range from the etching of sample cross-sections of bars, for the determination of inclusions and internal flaws, to the pickling of all bar stock, followed by sand blasting and close inspection to disclose such defects as roll seams and results of cold working. All forgings should be so inspected, to discover any surface defects, and the inspection may be carried still further by etching important cross-sections to determine the direction of grain or fiber resulting from the rolling or forging.

The more vitally important bolts, such as crankcase through-bolts, master-connecting-rod bolts and crankshaft bolts, should be given an additional heavy pickling, sand blasting and inspection after rough machining, to make sure that no internal defects were uncovered in the rough machining operations. No amount of final inspection can take the place of pickling and sand blasting, coupled with 100-per cent close visual inspection, as the action of the pickling acid followed by sand blasting will open up and make easily visible flaws that could not be detected visually in the finished part.

Guarding Against Localized Stresses

Inspectors must watch with especial care for tool marks on all surfaces which are not subsequently ground, as such tool marks result in concentrating the stresses at a given section and frequently cause fatigue fractures, which usually will not occur until the parts have been in service for a considerable time. We recommend grinding all surfaces, including lightening holes, wherever possible, to eliminate all tool marks, rather than to take the care otherwise necessary to eliminate all tool marks from machined surfaces.

Grinding operations frequently pay for themselves in savings on preliminary operations, in addition to the satisfaction of knowing that every possible precaution against fatigue failures has been taken. Engineers should remember, however, that specifying a 1/64-in.-radius fillet in corners of ground work requires that the grinding wheel shall be dressed for each finish-grinding operation. This results in loss of time and increased grinding-wheel consumption. Larger fillets are not subject to these difficulties to the same extent and are inherently stronger.

The same danger decrees that milled threads should replace die-cut threads on all important bolts and studs, as the minute tears unavoidably left at the root of die-cut threads introduce an element of uncertainty in service. Die-cut threads, when screwed into aluminum, also tend to re-tap the hole. Thread milling is one of many applications of machinery which is standard in the plant of the parts specialist, but would be special and but little used in the plants of most engine builders, because of insufficient volume of production.

Cylindrical-lapping machines are another example of this condition. It has been our policy, in case of subassemblies such as the valve lifter and its guide, to machine, grind and lap the parts in pairs, leaving a clearance of approximately 0.0002 in. between them for a running fit. Such a fit prevents throwing oil from a radial engine and practically eliminates wear of these parts.

Our practice is to grind the slots in all valve lifters and their guides, because of the distortion of the ears occurring during heat-treatment. We grind the cross holes, so that the pins will not distort the lifter at assembly. All valve-lifter guides made by our company are of carburized steel, of either 3115 or 2512 S.A.E. specification, and are soft on all surfaces except the holes. These parts can be made very strong and light by this method.

By furnishing the combined special-parts requirements of a large percentage of the aviation-engine builders of the Country, it is possible to provide equipment which would not normally be found in engine-building plants. The production of milled-thread bolts and studs and machine-lapped valve-lifter assemblies are typical of many services which the parts specialist can render.

Checking Heat-Treatment Is Important

Complete determination of the correctness of heat-treating operations is probably the most difficult of all inspection problems, yet no operation can so easily vary widely from the specifications laid down, and in no other operation can a slight variation from specifications so vitally affect the quality of the finished parts. For heat-treatment results of the highest type, dependence should be placed only in furnaces subject to the automatic control of recording pyrometers, frequently checked with indicating pyrometers, both pyrometer systems being maintained in condition by double-checking against a master standard.

Some indications of heat-treatment results are classified, approximately in the order of conclusiveness, although combinations of the various indications will be required for most thorough results, as follows:

(1) Investigation of hardness, by Rockwell, Brinell or scleroscope instrument, the Rockwell being preferred for steel

(2) Investigation of tensile strength, yield-point, elongation, reduction of area, and other physical properties

(3) Impact and fatigue tests

(4) Photomicrographic investigation and expert analysis of the results

Many aircraft-engine parts are carburized, and this work should be done by those familiar with the work and with the dangers involved. Unlike the conditions attending the carburizing of most ordinary work, the sections of aircraft parts are frequently extremely thin. Too deep penetration will result in a dangerous reduction of core strength, the extent of which cannot be determined without spoiling the part. Adequate checking of carburizing can therefore be made only partially, by including test samples of the thinnest sections and subsequently breaking these to determine the depth of case. Nothing in the way of inspection can take the place of the highest degree of skill and technique on the part of the heat-treatment management and operatives.

Non-Ferrous Metals and Alloys

Aluminums and bronzes, of the non-ferrous metals, are most in evidence. Both of these have their peculiarities. Aluminums are particularly sensitive to correctness of tool forms and settings and to heat-treatment. The bronzes—particularly the more recently used tin-lead bronze, which is ideal for certain purposes—are extremely sensitive to slight errors in analysis and are also subject to porosity. Detection of porosity should be made certain by rough-turning all parts and subjecting them to 100-per cent inspection before performing finishing operations. Frequently an entire lot of castings has to be rejected on account of porosity.

The alloy steels most frequently used are of the following groups: nickel steel, 2300 series; nickel-chromium steel, 3100 series; chromium steel, 52,100 series; and chromium-vanadium steel, 6100 series. This last-mentioned material has been found by us to possess ideal characteristics for certain uses, but it is extremely tough and stringy to machine. It is not readily obtainable from the mills, and appears to be more subject to seams and defects than the nickel-chromium material.

Sample Material Specification

EX-CELL-O AIRCRAFT & TOOL CORP., DETROIT, MICH.
Extruded Aluminum Bronze, Specification No. 236

Workmanship and Finish:—Bars must be sound, straight and of uniform quality; they must be free from laps, cracks, twists, seams, damaged ends and other defects. Any bar in which defects are revealed by manufacturing operations shall be replaced by the manufacturer, notwithstanding the fact that the bar has previously passed inspection. The full weight of the original material in the rejected bars shall be returned to the manufacturer at his expense.

Composition:—The material described is a homogeneous alloy of the following compositions:

Copper	89 to 91 per cent
Aluminum	9 to 11 per cent
Total impurities, not over	1.5 per cent

Physical Properties:—Bars up to and including 1 in. in diameter shall show the following minimum physical requirements:

Ultimate tensile strength	80,000 lb. per sq. in.
Elongation	15 per cent
Brinell hardness	174 minimum

Bars over 1 in. in diameter will be accepted on Brinell hardness alone, which must be 174 minimum on the surface of the bar. Hardness and tensile properties, if heat-treatment is required, shall be as specified on the drawings.

Identification:—The manufacturer's heat number must be stamped on every bar, slab or forging, if possible. Otherwise, small bars or forgings must be bundled and this information placed on the tag attached to the bundle.

The manufacturer must furnish the metallurgist of the Ex-Cell-O Aircraft & Tool Corp., Detroit, Mich., with a notice of each shipment, showing the chemical analysis, the order and requisition numbers and the number of pieces or weight of bars.

Rejection:—Upon rejection of material, the shipment will be charged back to the manufacturer and held at his risk for a reasonable length of time awaiting shipping instructions, the manufacturer to pay all transportation charges. It is understood and agreed that, at any time within three weeks after a shipment is rejected, the manufacturer may claim a portion of the chips used in making the analysis, for verification.

Its use is therefore attended by the necessity for increased vigilance in inspection.

Such a subject as this cannot be adequately covered in the time allowed, but I hope that this paper may at least drive home the consciousness that quality, safety and cost are relative and comparative terms and that, in spite of the savings incident to special equipment and experienced personnel, there is a limit beyond which costs cannot be reduced without compromises in quality and safety, which cannot be compromised without incurring incidental costs that may be appalling.

Equipment, experience, skill and above all a long and cherished record of conscientious achievement are the wings upon which the various branches of the industry must mount, be they airplane manufacturers, engine builders, or parts specialists.

THE DISCUSSION

CHAIRMAN EDWARD P. WARNER[2]:—Mr. Wise's suggestion that undue trouble and expense for the builder of parts will result from the use of fillets of abnormally small radius is a reminder which could be supplemented with many more of the same sort. The mere fact that a thing calculates out properly or looks pretty on the drawing or that it would be immune from service trouble when constructed is no guarantee that it is really a satisfactory design. Production, as a leading element in the industry, imposes upon the designer a responsibility of considering the production man, who has only comparatively recently become articulate enough to demand that the designer try to demonstrate his efficiency by getting out a set of drawings that will represent a machine which not only is

(*Concluded on p. 795*)

[2] M.S.A.E.—Editor of *Aviation*, New York City.

Friction-Coefficient Research

Discussion of Louis Illmer's Annual Meeting Paper[1]

THIS paper is a comprehensive survey of representative experimental research directed toward perfect-oil-film lubrication. With this as a basis, an endeavor is made to correlate empirically, by means of analytical research, the experimental results obtained by different investigators and to establish certain underlying principles common to all such tests in which the frictional resistance depends primarily upon fluid shear.

The conclusions arrived at are predicated upon a rather wide range of reported determinations, including most of the relevant tests that have found their way into friction literature during the last 50 years. In piecing together this accumulated experience, along the lines suggested some years ago by Prof. M. D. Hersey[2], a reasonably reliable and relatively simple method for predetermining certain kinds of friction losses has been deduced.

Discussers credit the author with having done a great amount of careful work on the subject and having produced a paper that is of value because it gathers together for reference the records of experimental work done all over the world. Several assert, however, that the equations presented are very difficult to understand and seem to complicate rather than simplify the subject.

The author is said to have handled his treatment almost entirely upon an empirical basis, and his equations are declared to be less readily usable than the rational equations that are already available. It seems to one discusser more effective to start with the proved scientific basis and develop on it the necessary extensions and modifications to make the results more readily applicable in practice.

Another discusser suggests that the author could better have shown how experimental results check with the rational theory based on the ZN/P factor, which is the product of the viscosity times the number of revolutions divided by the unit pressure. As the Otto and Diesel cycles are accepted as basic for comparison of engine performance, so the ZN/P diagram should become the basic method for plotting friction coefficients, as it is far simpler than the methods suggested in this paper and leaves the mathematical theory where it belongs as basic for the ZN/P relationship.

Replying to his critics, the author states that he used only algebraic equations throughout the paper and that the mathematical treatment is relatively simple. He points out wherein the ZN/P relation does not take into account certain variables that presumably control friction behavior and fails to check with certain experimental data, especially in the case of loosely fitted bearings. The merit of his contribution depends, he states, upon whether the key equation used was properly verified, and the correctness of the deductions summarized in this equation does not seem to have been seriously questioned. The analogy of the Otto and Diesel cycles does not seem pertinent, as the rational thermodynamic standard of performance really lies in the Carnot cycle, the equivalent of which, as applied to friction research, is most nearly approached by the author's Equation (12).

Professor Goodman, in a written communication, expresses the opinion that the author's work bridges the gulf between the research worker and the practical man and will be of great value to future workers in this field. The time, he asserts, is not ripe for even an approximate mathematical treatment and an empirical treatment of this subject seems to be the only method of dealing with it that is likely to lead to useful results.

DR. H. C. DICKINSON[3]:—I do not know whether I am capable of discussing this paper, which I have read with some care. It impresses me that the author has done a tremendous amount of very careful work on the subject and has arrived at some conclusions which, perhaps, may be useful if applications can be made. On the other hand, it appears that he has tended to bring chaos out of order rather than order out of chaos; that is, he has quoted 25 or 30 empirical equations elvolved by others, but none of which includes the latest systematic work on the subject.

The author refers, for instance, to some work of 15 years ago by Professor Hersey and some work of Mr. McKee's of more recent date, but has handled his treatment almost entirely on an empirical basis. The results are reduced to equations or formulas, but all of these equations, so far as I can determine, are empirical rather than rational. It is possible, to be sure, that some of them may be reducible to the rational equations, but I did not have time to prove it. However, if they can be so reduced, they are considerably more complex in form and less readily usable than the rational equations that are already available.

Some ten or more years ago, Professor Hersey and others established the fact that there is a single complex factor which enters into the lubrication problem and which is rather clearly determined on a rational basis; namely, the factor ZN/P, this being the form in which Mr. Illmer has used it. This factor may be expressed in different units, but in any case is the product of the viscosity times the number of revolutions, divided by the unit pressure. This factor can be used in

[1] Published in S.A.E. JOURNAL, January, 1930. The author is development engineer and patent attorney for the Brewer-Titchener Corp., Cortland, N. Y., and is a member of the Society. In the absence of the author, the paper was read at the meeting by J. P. Stewart, automotive research engineer, Vacuum Oil Co. The abstract accompanying the paper is reprinted herewith, supplemented with a brief summary of the discussion herewith published.

[2] See *Transactions of the American Society of Mechanical Engineers*, vol. 37, 1915, p. 197.

[3] M.S.A.E.—Chief, heat and power division, Bureau of Standards, City of Washington.

various forms, and Mr. Illmer has used in his paper something that resembles it, but I am not sure whether it is exactly the same thing.

Going back to the equations of Sommerfeld, which were not very readily handled until this factor was isolated by dimension analysis, it turns out that the relation between the coefficient of friction and these three quantities can be calculated on a purely theoretical basis. Those calculations, of course, being theoretical, involve among other things one assumption in particular which obviously is only an approximation; namely, the absence of end leakage. In other words, they involve uniformity of pressure distribution from end to end of the bearing.

It was not known how closely those calculations might agree with carefully performed experimental work until the work done some two years ago by Mr. McKee served to clear up that situation. Mr. McKee undertook the most accurate and systematic piece of work with which I am familiar to cover that field. He made a comparison between the theoretical values from the Sommerfeld equation, based on the ZN/P relationship, and the actual values determined by measurement for different ratios of length to diameter and different clearance ratios.

ZN/P Relationship a True Characteristic

Three of the most important factors that enter into analysis of journal-bearing performance are the length-diameter ratio, the clearance ratio and the ZN/P relationship. So far as these are concerned, McKee's experimental work showed that without any queston a single bearing will give the same value of f for a given value of ZN/P, no matter how that result is derived; that is, by varying the viscosity, the speed and the load in any way so as to get the same numerical value for ZN/P. The f, ZN/P curve therefore is a true characteristic for any particular journal bearing. This is shown experimentally within the limits of accuracy of work, and those limits are very close indeed; they represent some of the most accurate work I have seen on problems of this sort.

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Now, for any different ratio of length to diameter or different clearance ratio, we of course get a different ZN/P curve. Mr. McKee's results showed, for instance, that for values of L/D which are between $\frac{3}{4}$ and 1, that is, for nearly "square" bearings, the experimental results followed very closely the Sommerfeld theoretical equations. The results are very close together, almost within the limits of accuracy with which the clearance can be measured, especially for the smaller clearances that become very difficult to measure; in fact, it was impossible to measure the small clearances, in the order of 1 to 1000 and 1 to 2000, closely enough to determine whether or not the theoretical and the experimental results differed.

As soon as one departs from a rather narrow limit of L/D, for instance, using a bearing of the order of one-quarter or one-third the length, the factor of end leakage obviously comes in, and one would not expect the theoretical equations to hold so accurately. The experimental results show very clearly that the departure from the theoretical value of the friction is what is expected in this case.

When one goes to longer bearings, another factor comes in that throws the results off; namely, shaft deflection. One cannot have a very long bearing in which clearances are of the order of a few millionths of an inch without some shaft deflection, so that the longer bearings, having L/D ratios of 1.5:1 and longer, begin to show an increasing departure from the theoretical relationship. The result is that the whole problem, instead of being in such a confused state as indicated by Mr. Illmer's paper, seems to me to be in a very reasonably satisfactory state so far as agreement between the theoretical and the experimental results is concerned. Rather than delve into the analysis of a large number of complex empirical equations, it seems to me that it would be much more effective for one to start with the sound and proved scientific basis, so far as it goes, and develop on that basis the necessary extensions and modifications to make the results more readily applicable in practice.

GEOGE A. ROUND[4]:—After several hours' study of Mr. Illmer's paper, I am in thorough agreement with Dr. Dickinson's comment to the effect that the author has complicated rather than simplified the subject with which he is dealing. For all practical purposes I believe that the ZN/P formula is adequate and that most of our troubles come from conditions not connected with the factors discussed by the author in his paper.

Simpler Method Should Become Basic

ARTHUR E. NORTON[5]:—This paper will serve as a valuable work in gathering together in one place for reference the records of experimental work done all over the world during the last 40 years, beginning with the famous work of Tower.

I am sorry that the author did not relate his work more closely to the accepted and relatively simple theory on which the subject of lubrication is based. As I see it, he has attempted to find very elaborate empirical equations which will fit plotted curves resulting from these experiments of many years. I would suggest that he could have better shown how these results check with the rational theory based on ZN/P which Dr. Dickinson has mentioned.

It seems to me that we have a very good analogy from another field of engineering in which the method I propose has yielded far-reaching results. I refer to the development of the steam, gas and oil engines. The early tests were made with many good results, but real progress was made only by studying the test results as related to the theoretical cycle. Just as the Otto and Diesel cycles are accepted as basic for comparison of engine performance, so the ZN/P diagram should become the basic method for plotting tests of friction coefficients. It is far simpler than any of the methods suggested in this paper and leaves the mathematical theory where it belongs as a basis for the ZN/P relationship.

[4] M.S.A.E.—Assistant chief, engineering division, Vacuum Oil Co., New York City.

[5] Associate professor of mechanical engineering, Harvard University, Cambridge, Mass.

I think the author takes too seriously the Goodman tests referred to under Item A 21 in his paper. Anyone who reads the report of these tests will find that Mr. Goodman himself was very modest and uncertain about them and that one of the discussers of his paper had the courage to point out some very definite weaknesses in the tests. I am quite sure, in that case, that even the author of the tests would hardly wish to have them put on record as any basis for mathematical deductions like those made by the author of this paper.

Mr. Illmer emphasizes a very important point when he refers to contact factor and points out that a continuous (360-deg.) bearing rarely acts as a full bearing, but will preserve a film only over an arc less than 180 deg. This is called a "partial bearing." On this point I would mention the paper by Karelitz on Performance of Oil-Ring Bearings[6] recently delivered before the American Society of Mechanical Engineers. Karelitz finds many interesting things about the performance of these bearings and the extent to which the rational theory is verified. It is a good guide for the design of many partial bearings.

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As to bearing tests in general, it seems to me that, while the earlier tests were very valuable in their day, they can never be adequately correlated, because no common technique had been developed. I believe that tests like those of Hersey, McKee and others at the Bureau of Standards are more useful in determining a code of design than those of the past.

CHAIRMAN L. P. KALB[7]:—I was greatly pleased to hear these men of established mental attainments admit that some of the formulas were a little deep for even their comprehension. I thought it my duty to understand a little bit about this paper, as I was going to officiate at this meeting, but I must admit that my efforts to do so were unsuccessful.

S. M. UDALE[8]:—Attention is drawn to the paper by Prof. G. E. Charnock on Bearings for Line Shafting[9] in which he says:

> Assuming the viscosity of the lubricant to remain constant, it has been found that, within certain limits, the coefficient of friction for a given pressure is proportional to $\sqrt{L}$, L being the length of the bearing. Some experiments carried out by Brown, Boveri & Co. were first quoted in confirmation of this. A bearing originally 4.93 in. in diameter and 7.87 in. long was reduced by stages to 2.5 in. in length, causing the pressure to rise from 63 to 210 lb. per sq. in. of projected area. At a speed of 3000 r.p.m., equal to a rubbing velocity of 3879 ft. per min., the coefficient fell from 0.018, with the original length, to 0.009 with the reduced length.

This confirms what is generally known, that a large number of small bearings is better than a few larger ones, quite apart from the mechanical advantage in a crankshaft, for example.

Attention is also called to a paper by Francis Hodgkinson on Journal Bearing Practice[10] which resulted in a letter[11] from H. Brillie, of Paris, which shows the relation between the point of minimum oil-film thickness and maximum oil-pressure as determined by revolutions per minute. These references to Charnock, Hodgkinson and Brillie are well worth reviewing.

The Author Replies to Criticisms

LOUIS ILLMER:—It is not strange that the chief criticism of this paper should come from those who are wedded to the ZN/P theory of friction. I am at a loss, however, to understand just why the paper under discussion should have been found so difficult to interpret, since only algebraic equations have been used throughout. From a mathematical standpoint, therefore, it would appear that the treatment has been kept relatively simple for a paper of this character. Possibly some of the more vital issues may have become buried in an effort to put over too much material. As an instance, the presentation might have been clarified by the omission of all the viscosity subject matter and that having to do with the nonessential S_o and K_o factors.

In contrast, attention is directed to the the reference cited as No. 14, entitled, The Theory of Lubrication. This article professes to be an elementary exposition, although based primarily upon differential equations which but few engineers are able to comprehend. It is frankly conceded that several important assumptions had to be made in order to be able to solve the equations in question. Hence it is not unreasonable for engineers to doubt whether these theoretical findings are tenable, particularly when the results do not consistently check with actualities. Such speculative mathematical deductions do not necessarily assure a sound premise; nevertheless we are asked to accept unqualifiedly the ZN/P relation as the crowning expression of the prevailing theory of lubrication.

Deficiencies of the ZN/P Relation

I preferred to build up my technique along independent lines, first, by fixing from well authenticated experimental data the more outstanding variables that presumably control friction behavior. As explained in the text, such procedure led to the need for employing a P/S basis rather than the simple P value; it also required the evaluating of the exponents r, m and n. As shown in Table 1 of the paper[12], all of these exponents do not invariably assume a fixed unity value, but are more likely to fall far below unity. The fact that the ZN/P relation makes no provision for a P/S correction indicates that the friction coefficient remains fixed irrespective of the degree of contact S, but this assumption is not in accordance with Fig. 14[a]. Such findings alone raise the question whether the alleged rational ZN/P relation is actually a valid one; also whether my procedure is so wholly empirical as the commentators try to make out.

The variables μ, V and P alone are not adequate to determine friction behavior. It is found necessary to

[6] Presented at the annual meeting of the American Society of Mechanical Engineers, Dec. 2 to 6, 1929.

[7] M.S.A.E.—Assistant chief engineer, Continental Motors Corp., Detroit.

[8] Patent attorney, Holley Carburetor Co., Detroit.

[9] See *Engineering*, Dec. 20, 1929, p. 801.

[10] See *Engineering*, Nov. 22, 1929, p. 690.

[11] See *Engineering*, Jan. 3, 1930, p. 22.

[12] See S.A.E. JOURNAL, January, 1930, pp. 70 and 71.

[a] See S.A.E. JOURNAL, January, 1930, p. 79.

take into account still another factor in order to arrive at a reasonably consistent result that will permit of a wide-range application. This missing factor, as embodied in equation (D13[b]), leads back to the fit factor which, for convenience, was allowed for in the paper in terms of the coefficient n^3. A loosely fitted journal invariably operates with a relatively low n value and it is not improbable that this exponent n is intimately associated with the corresponding contact factor S and that this in turn indirectly measures the relationship between β/δ.

Rather than speculate on the reason for the need of the cited correcton, I followed the more practical procedure of merely tying up with one such factor that would afford a suitable working basis for predetermining friction coefficients. Furthermore, it is not seen wherein the ZN/P relation makes proper allowance for extremely close-fitted journals, whereas the herein advocated $K\sqrt{D}$ factor as fixed by Equation (D4) prescribes a critical value beyond which the resulting friction coefficient is augmented instead of continuing to decrease indefinitely with fit closeness.

Does Not Check with Experimental Data

The cited ZN/P factor also fails to check with certain of the generally recognized experimental data that were rather thoroughly analyzed. This conclusion may be verified by the supplemental plots designated as Figs. 19 and 20 herewith. Fig. 19 depicts the Lasche tests (A6 in Table 1) as run under different speeds while the pressure and viscosity remain constant. In accordance with the ZN/P relation, the plot expectation is a straight line as drawn in the broken line, while the results actually follow the contradictory full-line trend. The same inconsistency can readily be demonstrated by resort to the Tower Tests (A1). Referring to Fig. 20, it will be seen that there is no simple and direct relationship between the ordinate and obscissa of the respective Tower curves designated as olive and sperm oils. In each of these tests, the viscosity μ was kept constant throughout and the variable placed solely in the velocity or N factor. Not only is no direct proportionality found with respect to the factor N, but the resulting friction coefficient between these two oils actually increases in a square-root relation rather than directly with viscosity μ.

LOUIS ILLMER

Turning attention now to more loosely fitted bearings, as in Fig. 21, this plot is representative of the Stribeck tests (A5), and it will be seen that the divergence from the ZN/P theory is even more marked. In accordance with this theoretical expression, all of these Stribeck curves should fall upon a single straight line heading obliquely upward from the zero axis, but this is decidedly not the case. A substantially similar plot can be obtained by using Stribeck's (A4) tests, wherein all factors except velocity are likewise kept constant.

Even when the ZN/P values do define a sloping straight line, the direction of the line does not usually head toward the origin of the graph and, accordingly, does not prove a direct proportional measure for the accompanying friction coefficients. As is clearly indicated by Fig. 22, any family of exponential curves, when carried sufficiently far from their origin, plot up into a series of substantially straight lines. The discrepancies discussed in the cited supplementary curves are thought to constitute sufficient reason for not following blindly the alleged rational fluid-friction theory.

Key Equation Accounts for Essential Factors

No serious fault seems to have been found with the correctness of the deductions embodied in the summarizing Equation (D13). The objections raised on the ground of complicated treatment may be answered in part by pointing out that the cited equation naturally reflects the taking into account of additional essential variables required to determine friction behavior.

With regard to the need for coordinating collective experimental data, the following quotation may be pertinent: "A fact is not necessarily valuable because it is a fact. It gathers significance only when it is compared with some other fact, and, if it is not comparable. it is not significant." The merit of the present contribution depends upon whether the key Equation (D13) has been properly verified.

Replying more specifically to certain of Dr. Dickinson's comments, reference is made to my statement included under the heading Concluding Remarks. In this I did not claim to have brought order out of chaos, but, rather, called attention to the need for so doing.

With reference to further utilizing the McKee experiments, these are largely reported upon the combined ZN/P basis and as such are wholly useless for present purposes. Also, as far as could be determined from the given data, the McKee tests operate with a circumstance constant C_2, that is greater than unity. Request was made upon the Bureau of Standards for the original test logs, but, owing to an apparent unwillingness to cooperate, the author was unable to secure the solicited information. Such rather unscientific mode of reporting friction determinations has lately become the vogue in this Country and is the one outstanding reason why more of our own friction tests could not be utilized. Fortunately, this vogue has not met with recognition in European literature, in which no similar attempt is made to obscure the test readings.

That the advocated ZN/P relation is not all-embracing is rather clearly indicated by the corresponding wide range of constants that must be prescribed for essentially identical services. In the reference cited in my paper as No. 15[c], there are listed ZN/P values ranging from 2 to 5 as applicable to Diesel-engine crankpins; for stationary gas-engine crankpins the constant is given as 15, while marine engine crankpins are listed as working with a constant of 20. In reality, the underlying principle of bearing design is practically identical in all these cases and hence should naturally conform to a common formula constant.

The reference No. 15 further gives a ZN/P value of 100 to 200 for stationary steam-turbine bearings and 1500 to 3000 in the case of small DeLaval turbine bearings, and even carries this so-called constant up to 10,000 for cotton-mill spindles. The fundamental

[b] See S.A.E. JOURNAL, January, 1930, p. 78.

[c] See S.A.E. JOURNAL, January, 1930, p. 83.

Equation (D13), as developed in the paper, calls for no such preposterous range, but rather specifically fixes upon a single concrete value for the friction coefficient under any given operating conditions.

In view of the foregoing, it appears that this particular art has not yet reached such a high state of perfection as Dr. Dickinson's comments would indicate.

Otto and Diesel Cycle Analogy Not Pertinent

Referring to Mr. Norton's comments: Professor Goodman's test designated as A21 checks up reasonably well with deductions from other loosely fitted journals such as the Stanton tests (A20) or the Fig. 9 data reported by Stribeck. The casting of doubt upon this particular experimental work is not well founded, particularly since Professor Goodman has long enjoyed world-wide recognition as an authority in the field of friction research.

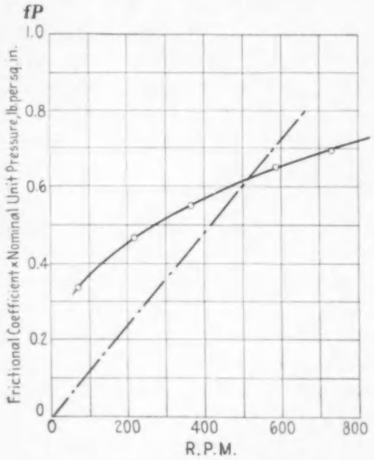

FIG. 19—LASCHE DATA PLOTTED IN $f\ P$ VERSUS R.P.M.

Curve Based upon Fig. 29 from Lasche in *Zeitschrift des Vereines Deutscher Ingenieure*, Dec. 13, 1902, p. 1881. Viscosity μ=Constant at 122 Deg. Fahr. Pressure P=Constant at 93 Lb. per Sq. In.

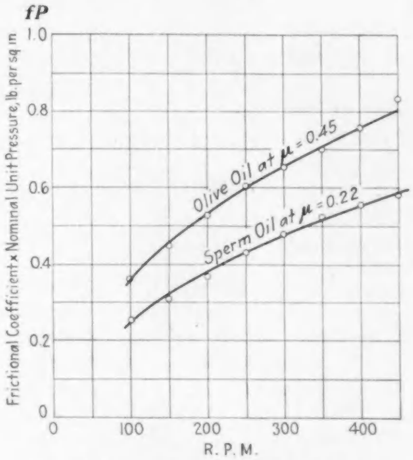

FIG. 20—TOWER DATA PLOTTED IN $f\ P$ VERSUS R.P.M.

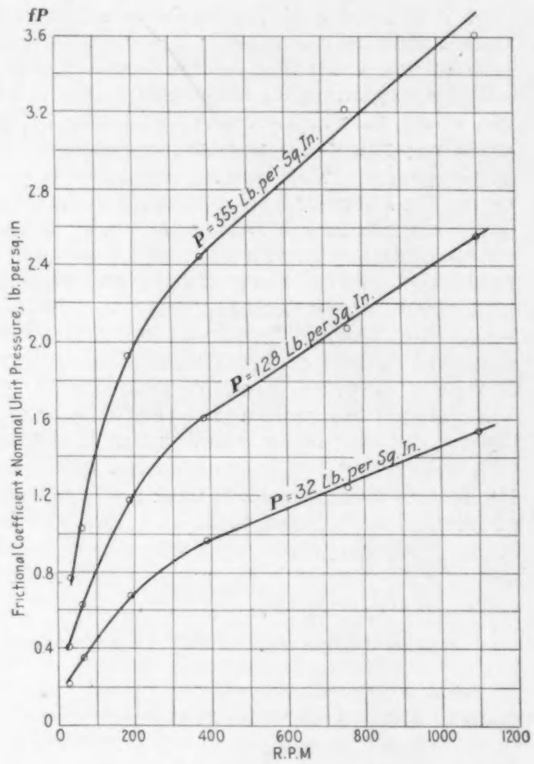

FIG. 21—STRIBECK DATA PLOTTED IN $f\ P$ VERSUS R.P.M.

Curves Based upon Valves Given in *Zeitschrift des Vereines Deutscher Ingenieure*, Sept. 20, 1902, p. 1436. Viscosity Constant at μ=1.95

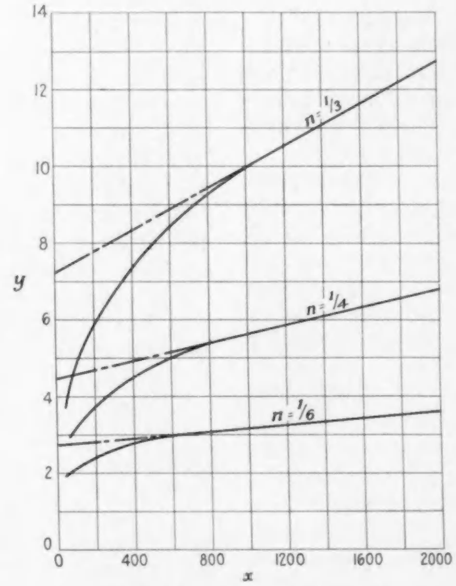

FIG. 22—EXPONENTIAL CURVES: $y = x^n$

As Indicated, Any Family of Exponential Curves, When Carried Sufficiently Far from Their Origin, Plot up into a Series of Straight Lines

The reason for having to rely rather heavily upon earlier tests has in part been explained. It is questioned, however, whether the more recent experimental contributions present anything more trustworthy than the classic Tower and Lasche friction tests. While having overlooked some factors that bear upon the results, these missing data can usually be assumed with a reasonable degree of certainty of their accuracy.

The cited Karelitz test appeared too late to receive consideration, but some of his earlier reports (B4) have been taken into account. The latter present the findings in terms of temperature rise rather than in more positive terms that would allow of evaluating the accompanying friction coefficients.

It is not apparent wherein the cited analogy to the Otto and the Diesel cycles is pertinent to the issue, since the rational thermodynamic standard of performance really lies in the Carnot cycle. As applied to friction research, the equivalent of the last-named cycle is most nearly approached in Equation (12) of the paper. An illustrative criterion more apt than any heat-cycle basis is furnished by the long-column formula, wherein the introduced constants are generally fixed by appropriate tests. That, in a sense, is the method that in the paper has been applied to friction research.

Professor Goodman's comment is to the effect that the time is not ripe for formulating a non-speculative mathematical treatment for predetermining fluid-friction coefficients. The failure of the ZN/P theory to meet expectations falls back upon end leakage as an outstanding disturbing factor, whereas it has been shown herein that when a bearing is kept in flooded condition the effect of the end leakage should be substantially compensated by the use of a more than ample supply of lubricant, that is to say that any reasonable restriction or loss on part of such supply is not likely to bring about a marked increase in the friction coefficient.

It is evident that the fluid-friction laws have not yet been adequately explored; after the effect of all controlling variables has been accurately determined from reliable experimental research by empirical methods or otherwise, it will be made far easier correctly to theorize regarding the underlying reasons for the resulting behavior. The bearing designer is primarily concerned in knowing *how* a bearing is likely to act under given circumstances rather than *why* it does so. Notwithstanding the contrary views expressed by my commentators, the present attempt at empirical coordination should prove decidedly helpful in leading toward a sound and workable rational solution of this problem that will closely conform with expectations as applied to widely different bearing practices.

Finally, it is repeated that the primary aim was to lay down a suitable basic equation for predetermining friction coefficients for bearing-design purposes. Whether the results would conform with any preconceived theory of lubrication did not enter into this problem. Merely being different from the traditional apparently furnishes reason enough for adverse criticism of the present mode of treatment.

Paper Represents a Valuable Piece of Work

PROF. JOHN GOODMAN[13]:—Almost every successful piece of engineering research work has to pass through three distinct phases before it benefits the engineering world. First, the research worker investigates the matter by either a mathematical or an experimental process. Second, the scientifically minded practical man digests the results obtained by the research worker and puts them in such a form that they can be appreciated by designers. Third, the practical engineer applies the research worker's discoveries to useful purposes. Each plays his own part, but it very rarely happens that one man can play the rôle of all three.

The author of this paper has evidently spent much time and labor in attempting to reduce to order a great variety of friction tests made under many different conditions and has thereby bridged the gulf between the research worker and the practical man. In this work he has shown judgment and skill; in the writer's opinion his work will be of great value to future workers in this field.

Mathematicians may find fault with the author for not following along the lines of accepted mathematical theories of lubrication. In my opinion it is well that he has not done so, because everyone who has done experimental work on the friction of bearings is well aware that theory and practice do not agree, largely because the assumptions on which the mathematical theory are based are far from the truth and highly important disturbances are ignored because of their extreme complexity.

In one instance the writer of a paper on the Mathematical Theory of Lubrication calculated that there was a vacuum of 2700 lb. per sq. in. on the "off" side of a certain bearing; even such an absurdity did not apparently shake his faith in the theory. Another mathematical writer frankly admits that it is almost impossible to calculate the effect of end leakage of oil from a bearing, yet he agrees that end leakage profoundly affects the thickness and distribution of the oil film, the attitude of the bearing and the friction.

Subject Cannot Be Treated Mathematically

In the light of such shortcomings in the theory of lubrication, it is evident that the time is not ripe for even an approximate mathematical treatment of the subject. Hence, until these disturbances can be dealt with by rigid mathematical analysis, an empirical treatment of the subject seems to be the only method of dealing with it. Even the simple relation used by the author, $f = \frac{\mu V}{P}$, with suitable powers, is known to be only a rough approximation, since the friction does not vary merely as some power of the coefficient of viscosity when the velocity and pressure remain constant. The friction also depends upon the differential expansion of the bearing and shaft due to temperature changes, the thickness of the oil film, the rigidity of the bearing, whch always distorts under a change of load, and the end leakage of the oil from the bearing.

According to the accepted mathematical theory of lubrication, no wear takes place with a flooded bearing and the nature of the material in the bearing surfaces has no effect upon the friction, but every experimenter knows that this statement is untrue. Hence it appears that, for the present, an empirical treatment of the subject is the only one likely to lead to useful results and to help bridge the gulf between theory and practice. It is hoped that an empirical treatment of the subject will furnish data to serve as a useful guide to future mathematical workers.

The data given in Table I are most valuable and cannot fail to be of assistance to future workers. The author's equations naturally become somewhat complicated when he attempts to cover the conditions of working, but in my opinion the expressions will necessarily become much more involved before the theory is anything like complete.

The oil-pressure diagram in Fig. 15, taken from my paper, An Experimental Determination of the Thickness and Distribution of the Oil Film in a Flooded Cylindrical Bearing[14], does not profess to be accurate. It was marred by the fact that the bearing was tilted in order to get more pressure points of observation and was given mainly to point out the inaccuracy of such a procedure. The reason that the pressure falls so rapidly on the "off" side doubtless is due to excessive end-leakage of oil from the abnormally large clearance. Experiments now proceeding on a bearing with a much smaller clearance do not show this effect to such a marked extent.

In my opinion the author has done very valuable service by preparing his paper.

[13] University of Leeds, Leeds, England.

[14] See *Proceedings of the Institute of Civil Engineers*, vol. 226 1928, Part II, p 242.

The Necessity of Standardization in Canada

By H. D. Allee[1] Canadian Section Paper

THREE major forms of standardization that are applicable to the Canadian automotive industry and by which the manufacturers can profit are pointed out by the author, who explains how they can be applied. The fundamental form on which the second and third are predicated is the S.A.E. Standards.

Canadian car manufacturers can help greatly, the author asserts, by using standard parts to the maximum extent possible without sacrificing individuality of appearance, thereby reducing tool and other production costs of the parts makers. Minor peculiarities of design should not be forced by American parent companies upon their Canadian subsidiaries. Use of S.A.E. standard parts as substitutes for parts of individual design would simplify instead of complicating the service-parts situation.

The standards committee of the Canadian Automobile Manufacturers and Exporters Association could deal with the standardization of materials and fabricated parts with advantage to both the car manufacturers and the parts makers.

Concentrating the many small hand-to-mouth orders into large volumes of standard parts through sales-engineering organizations acting as distributors would smooth out the flow of parts, take care of peak demands and give parts makers a larger volume of business over which tool and production costs could be distributed.

Many angles of the problem are dealt with by the discussers, major ones being the effect of the export trade and the tariff, service parts for overseas, relative size of the market in Canada and the United States, domination of the engineering staffs of Canadian plants by executives of parent companies in this Country, unnecessary minor design changes, the higher cost of materials and the lower volume of production per machine per man in Canada than in the United States.

THE NECESSITY of standardization in Canada is obvious. Standardization has many ramifications that are economically sound and by the use of which the industry as a whole, especially in Canada, can logically profit. The major forms of standardization applicable are: first, S.A.E. Standards; second, unification of design and third, the universal marketing of standardized products. These have been placed in the order of their importance, and the third is predicated on the first two, and the second is predicated on the first.

Without doubt the rapid advancement of the automotive industry has been the result of free interchange of ideas and methods. The industry today has no mysteries or jealously guarded secrets. A splendid example of cooperation was exhibited when lacquer for automobile finishes was introduced. I believe that every prospective user of lacquer visited all users and learned from them the advantages, disadvantages and questionable features, with the result that within less than 18 months virtually all of the difficulties in the plant and trouble in the field had been overcome. Today lacquer is the standard finish and remarkably little trouble is experienced with it. No one user could have accomplished the ultimate in the same time. It was accomplished only by the cooperation of all.

S.A.E. Standards Reduce Cost

Such interchange of ideas has been fostered by two powerful organizations: the National Automobile Chamber of Commerce and the Society of Automotive Engineers. The former has promoted the interests of all motor-vehicle manufacturers at home and abroad and protected the industry from the stagnating effect of patent threats and litigation. The Society has performed an equally important part by establishing standards of quality of the various materials used by the industry and standard dimensions of parts and fittings. Consider the complications that would arise were it not for the screw-thread standards alone; the loss in dollars would be appalling were each manufacturer to use thread specifications differing from those of others. Still the industry has a long way to go before it takes full advantage of the S.A.E. Standards. A great deal can be done in Canada in the matter of standard steels alone. Standard spring steels are fairly universal, but forging stocks are not. In this connection the supplier of forgings might well do more concentrated selling and educating toward the S.A.E. Standards.

Experts in their own lines in the industry have disinterestedly established these standards, and time has proved the advantages of the standards established. Let us then immediately accept and utilize the standards that are constantly being adopted by the Society. Why should certain manufacturers pay premiums and tool charges for special lamp-sockets or head-lamp mountings when the S.A.E. has established standards for them?

The fact that the automotive industry in Canada is small necessitates that tool costs shall be spread over the largest possible volume of production. Standardization does this. Tool costs probably are the greatest single factor in slowing up the sale of parts in Canada and consequently the greatest impediment to the development of volume production by the parts makers.

Standardization without Loss of Individuality

The obvious advantages of adopting S.A.E. Standards lead us to the second point, the unification of design. Every manufacturer, particularly in Canada,

[1] Production manager, Studebaker Corp. of Canada, Ltd., Walkerville, Ont., Canada.

should have leeway to deviate from approved design to S.A.E. design. By this I mean that if the parent company uses a peculiar design, the Canadian subsidiary should be permitted to use an S.A.E. standard design as a substitute. The reaction to this assertion might be, "That would complicate the service-parts situation." This would be a fallacy; in fact, it would simplify service, because parts made to S.A.E. Standards are today procurable everywhere throughout the world, even in corner drug stores and 5 and 10-cent stores. The reduction in service-parts inventory is in itself a saving of no mean proportions when we consider that parts imported carry import duty as a considerable proportion of their inventory value.

Unification of design in Canada should be carried almost to the point that it affects individuality. Our sales departments would never consent to loss of individuality of appearance, nor do we want them to, but by smart design the advantages of unification can be obtained without loss of individualism. For example, we can mount a head-lamp door of individual design on a standard head-lamp body and standard mounting and obtain a distinctly individual head-lamp. Similarly, we can select a set of body hardware, nickel it and obtain a certain appearance, Butler-finish it and obtain a different appearance, or apply a little enamel here and there and it becomes a third or a fourth set of hardware.

To me, one of the most gratifying strokes along this line recently in Canada was the adoption by three large concerns of the same wire wheel. This wheel, when mounted on possibly a dozen different cars with different hub-caps, becomes in appearance 12 different wheels. In this instance not a single company in Canada could have absorbed the tool charge on this wheel, yet the combined volume was sufficient for the producer to absorb it himself as a standard.

Let us mentally assemble a complete car and consider *what can be standardized* and *what must be individual.* The parts that can be standardized are so manifold that it is easier to list the major items, which are: body shells, fenders, hoods, frames, engines and radiators. That this is the *whole list* is not as strange as it might seem. Note that the first four items entail heavy tool-charges in the form of dies that would not be practical for absorption in small production. Engines for certain makes of car may or may not be produced in Canada; radiators are. This means that by proper unification of design along standardization practice more of the car can be manufactured locally. That this is not done today is because of the lack of a common meeting ground for engineers resident in Canada. The S.A.E. functions only partly along this line and deals with abstract ideas instead of concrete details, except in the case of the Standards Committee.

What a Canadian Standards Committee Could Do

The Canadian Automobile Manufacturers and Exporters Association had a standards committee about two years ago, and it was the original intention of this committee to establish uniform standards for such common materials as steel for springs, forgings, frame stock and the like, but it did not consider standardization of the non-ferrous or fabric materials or the greatest possible source, which is in actual fabricated parts, such as steering-wheels, steering-gears, transmissions, propeller shafts, body hardware and trim, including curtain rollers, wind lace, buttons and fabrics.

The advantages of a committee functioning along these lines should be readily appreciated by the car manufacturer as well as the parts maker, and the few items listed above should suffice to start everyone interested in considering how he individually could profit by such action. I recommend and hope that such a representative gathering will be called together in the very near future to get under way on this most important function.

One thing which such a committee could do, if nothing else, is ascertain what parts are generally used and endeavor to increase the volume for the parts makers. This would enable the parts makers to reduce the costs so that the parts would be competitive. At present many of the parts and materials purchased in Canada are at a premium when used in cars having only 20-per cent tariff protection.

Sales Organization for Standard Parts Needed

The lack of a common ground brings me to the third phase of standardization, the necessity for the universal marketing of standardized products. To me, the lack of proper contact between parts makers and car manufacturers is the biggest single source of economic loss in the industry in Canada. This failure of the parts makers to operate through sales engineers or sales representatives limits their individual volume materially and has a stifling influence on the industry as a whole. At present the parts maker is contacting with the manufacturer direct, usually through some manufacturing executive. He neglects his own organization when he is absent and does not promote the best interests of his company when contacting. He is too prone to *buy* the idea of the man he is dealing with instead of selling that man on the standardized product. I claim that a producer is not a salesman and cannot function as such. He is too confined to a particular thing and does not have the vision to place a different article in a given place and have it function satisfactorily, both mechanically and esthetically.

I feel very keenly on this point because I am a producer and know there is more money in selling. Much of the reduction in material costs of cars in the United States has been brought about by keen sales-engineering firms taking a good standardized product and adapting it to volume sales to a large number of car manufacturers. The parts maker will doubtless question the advisability of incurring the cost of a sales agent, but when he considers his present sales expense and the potentiality of an increase in volume to lower his overhead, he will find that the small percentage charged by the sales agent is more than offset; in fact, he should be able to decrease his selling price. Doubtless many products have ceased to be used because of the lack of proper contact that would have maintained a market for them.

Small-Volume Buying Increases Cost

Another function of a sales-engineering organization, which directly ties into production, is the proper distribution of standardized articles. Every manufacturer is paying a premium because he is getting his material direct from the supplier and in small volume. Why not bring all these small lots into a community in large volume and redistribute locally? This is particularly applicable to small parts such as nuts, bolts, screws, hood locks, mirrors, hardware and similar items. We all know the saving effected by shipping in carload

lots instead of l.c.l. or transport, but our small volume will not permit us to realize the saving. A sales-engineering organization can handle the distribution so as to obtain carloads and pass along this advantage to the purchaser.

Changes are being made constantly and will continue to be made as long as the industry progresses. The possibility of change means that the manufacturer who is working on a sound basis keeps his inventory as small as possible, which means buying in small quantities and making frequent and rapid turnover. It has been stated that the profits to be realized in 1930 are entirely within inventory control, and at no time in the history of the industry will this one thing be the crux of the situation to as great an extent as it is at present.

Distributors Could Equalize Flow of Parts

A middleman distributor can control the volume of the parts makers economically and at the same time satisfactorily accommodate the hand-to-mouth buying of the car manufacturer. Hand-to-mouth buying has been forced upon us again by necessity, and let us hope it stays this time, because it is economically sound. The success of chain stores is the outstanding example of profits made in quick turnover. Needless to say, a large number of customers buying in small lots would level production for the parts maker and allow him to realize a saving through increased efficiency.

The industry in Canada is ideally suited for the functioning of sales engineering because there are only two centers of importance: Toronto and the Border Cities. Think what it would mean to the manufacturer in Windsor to be able to call in a sales engineer, outline a problem and receive the answer in a standardized product all in the course of an hour.

Let us consider the other angle, that of distribution. A manufacturer's production is increasing and necessitates the advancing of schedules; the parts makers' sales organization is advised of the urgency and has the material coming through that can be immediately diverted; thus the manufacturer is taken care of temporarily and the parts makers take up the slack gradually and efficiently. Under present conditions, the parts makers work overtime for a day or two and then slow down and the car manufacturer either draws a carload or pays the cost for quick delivery. This is unnecessary and an economic waste.

We must not lose sight of the fact that the automotive industry is the acme of versatility, and that anything done to maintain its versatility without loss is a step in further progress. A sales-engineering organization can do just this.

The automotive industry in Canada has been in a condition of "feast or famine" because of varying volume and weather conditions. We cannot do anything about the weather, but we can do a great deal about the volume. Standardization and distribution are perfect creators and regulators of volume. Let us then make the utmost use of them.

In closing, I want to emphasize four points for your consideration; namely: (*a*) better contact between manufacturers and parts makers through sales engineers; (*b*) a forum for manufacturers; (*c*) greater volume and lower cost on parts that do not affect the individuality or mechanical functioning of the car and (*d*) better distribution, to reduce transportation costs on parts purchased in small volume.

THE DISCUSSION

CHAIRMAN R. H. COMBS[2]:—From my experience as an accessory manufacturer I agree fully with Mr. Allee in all his findings. Up to 1917 I was in the United States, where we dealt with a population of 120,000,000 persons. In 1917 I came to Canada and since then have been forced to change my ideas and think in figures of 8,000,000 or 9,000,000. I have got to the point today, when I think at all of volume, that I always multiply my thoughts by 15 and try to imagine what the man across the border is thinking. One of our greatest difficulties in the automotive industry in Canada is that we are not independent enough of the 120,000,000 figure. Our 8,000,000 is submerged under the 120,000,000 figure, because of the more or less dominating position of our American parent companies. Most of our Canadian companies are the offspring of the larger companies on the other side, the effect of which is that most of us who are operating over here are rather submerged because of our smallness. I think we are hampered by the fact that the man we report to generally talks Canadian and thinks American; he talks 8,000,000, but thinks 120,000,000.

If we could by any means induce the car builders here to use the same shackles or brake-linings or bolts, it would help a great deal. As it is, the small volume in any part is divided among five or six suppliers, and their overhead cannot go up the smoke-stack; it is too heavy. Right there is the crux of the matter.

C. E. TILSTON[3]:—What Mr. Combs said is very true. One important factor that enters into production, so far as we are concerned, is that approximately one-half of our production is for export. We must work to American specifications and cannot use Canadian batteries on export cars. The same thing applies in a number of cases. A Canadian company makes a wheel that is not the same as the American wheel. How would Mr. Allee handle a problem like that in export?

H. D. ALLEE:—We regard a car exported from Canada as a Canadian car and handle all the service for cars produced in Canada. We pack parts that are peculiar and ship them from Walkerville, Ont.; parts that are common we ship overseas direct from the United States. We build the same car for export as for the Canadian trade.

MR. TILSTON:—We have a problem right now with wheels. Unfortunately, we do not handle service from the Canadian plant and, if we ship service parts from the American factory and cars from Canada, we get into a confused condition.

MR. ALLEE:—Our serial numbers govern the whole matter. If an order goes to the States, it is checked off by the serial number, which involves a delay of only about a day.

MR. TILSTON:—Do you export into the same territory as the American company?

MR. ALLEE:—Yes.

[2] M.S.A.E.—President, Prest-O-Lite Storage Battery Co., Ltd., Toronto, Canada.

[3] M.S.A.E.—Chief engineer, Willys-Overland, Ltd., West Toronto, Ont., Canada.

MR. TILSTON:—Do all your foreign dealers carry stocks of both Canadian and American parts?

MR. ALLEE:—They do if they have cars produced in the two countries.

MR. TILSTON:—Does that not lead to extremely large stocks?

MR. ALLEE:—Yes, that is so, but only of parts that are peculiar. In the case of wheels the dealers have to carry a double stock.

MR. TILSTON:—That means that the export corporation has to tie up so much more capital.

How the Export Field Is Divided

CHAIRMAN COMBS:—What is the dividing line, Mr. Allee, between what parts go from Canada and what parts go from the United States?

MR. ALLEE:—Some car models we do not build in Canada, and parts for them are shipped from the United States.

A MEMBER:—We have the same condition. We export into the same territory as the American company and in general we export where we have preference; that is, the British preferential tariff governs the situation. In a number of cases an order comes in that we cannot fill and it is filled from the United States. That leads to a direct comparison, and we find that, rightly or wrongly, if anything goes wrong with a car shipped from the smaller Canadian plant and the part that fails is of Canadian origin, that is criticized rather than the design or the operator.

CHAIRMAN COMBS:—We find that condition ourselves. We do not mix the fields in our line. We have a definite territory for everything shipped from Canada and others for everything shipped from the American plant. Sometimes we ship into a field in which our Canadian product will not meet with the same approval as the American product did, and sometimes the American plant ships into a field in which its product does not get approval. Everything we market in Canada is produced in Canada.

How Car Builders Increase Parts Cost

G. J. MONAGHAN[4]:—Some parts makers say that the automotive trade as a whole is to blame because the manufacturers cancel orders. We have to make a set of dies and after producing a few parts the design is changed a little here and there, getting into individuality again. I think the engineers of the automotive plants could help standardization considerably by avoiding the changes that are constantly being made and that hamper the parts makers.

As for schedules, some plants I call on one day will be working 24 hr. and the next time they are shut down and I am told that the automotive manufacturers have shut down on deliveries. If the automotive people could even the schedule, they could ultimately reduce their own costs. If they were to go through the parts plants, they would realize the cost on the small parts, the little punch pieces. It takes time for the parts makers to get stocks of material. They have not much capital; they have to order stock for a rush shipment, then set up the machines to run a little batch now and then, instead of having a constant stream. I am not a parts manufacturer, but it seems to me that if the manufacturers would cooperate with the parts makers they would help themselves.

MR. ALLEE:—That is a very good point. The farther one goes from the American car the less subject he is to change. Right now we are not touching changes on the American cars; we are far enough away so that we can forget the little knick-knacks that are put on in the United States. As you go farther away, that condition improves.

CHAIRMAN COMBS:—Less than a week ago one large American motor-car manufacturer with whom we have a contract made some little change in design that would reduce production costs by 5 or 6 cents. Someone thought that by offsetting the posts of the storage-battery the connecting link would be shorter. It left the battery as efficient and saved some lead. The Canadian automobile manufacturer wanted to pass this change in design along to me because our American company gave it to the American parent car company. I said, "It will cost me 20 cents a battery for dies and tools and you will take this battery for one year and then want to cut off another cent's worth of lead. I will be giving you 4 cents and it will cost me 20 cents per battery." That is the trouble with Canadian costs today, and when the Tariff Commission meets, we go to Ottawa and try to tell the commissioners why we cannot reduce Canadian costs.

Car Builders Pay for Unstandardized Practices

N. P. PETERSEN[5]:—Mr. Allee's paper has described a condition and suggested a remedy. I cannot help voicing my feeling that he is one preacher who really practises what he preaches. If the majority of the other manufacturers would see eye to eye and treat the parts makers in the same way that he has treated some of the problems we have had to face, there certainly would be a large saving. The question is, How can we get them to see the force of his contentions, and how can he and some other persuasive salesman sell the idea to a majority of the car manufacturers?

CHAIRMAN COMBS:—The Canadian motor-car manufacturer is punished in two ways for lack of standardization. One is in his inability to obtain the right price on parts, and we will couple with that the shrinkage of supply. Every Canadian manufacturer today is unable to get sufficient material when he runs into peaks. During three months last year every motor-car manufacturer was scrambling to get enough material to keep his factory going, with the result that when sales failed in August and September, he was loaded with parts and materials that were delivered when the mills and machines were going at high speed.

If one company would use exactly the same things as the other large Canadian companies are using, barring little individualities that might be added, the parts maker would have a chance to reduce costs and be able to keep his plant going on a little more level basis and carry a little inventory himself.

About one-half of our Canadian production is for export. Most manufacturers have two Canadian sources of supply, and with the United States source, the business is divided three ways. Last year 105,000 cars were exported out of a total of 263,000 produced; that is 40 per cent export and 60 per cent domestic business. We know that the major volume of the material that went into the 105,000 exported cars was American products because of the drawback provision of the Canadian tariff. The Canadian Government gets 1 per

[4] Sales manager, Canadian Raybestos Co., Ltd., Peterborough, Ont., Canada.

[5] M.S.A.E.—Works manager, Canadian Acme Screw & Gear, Ltd., Toronto, Canada.

cent from the duty and we get a little out of the assembly; but virtually all the export business is American.

Production Costs High in Canada

A. H. WOOD[6]:—Is it not a fact that some of the Canadian parts manufacturers have little or no engineering departments to study the needs of the car manufacturer in their particular line? I have in mind the placing of an order for a very large number of major units, but when the samples came through, they did not measure up to the required specifications, with the result that the order went back to the States.

CHAIRMAN COMBS:—A Canadian manufacturer, whether he is building storage-batteries or tie-rods or brake-linings or hubs or what not, cannot be expected to compete when he gets probably only one-thirtieth of the business because of the export trade, which excludes the Canadian manufacturer. Many materials are higher in Canada than in the United States. Every manufacturer in Canada, whether he is making motor-cars or wheelbarrows, is confronted with the same condition.

[6] M.S.A.E.—Technical department, General Motors of Canada, Ltd., Oshawa, Ont., Canada.

Another thing that interferes with Canadian producers is that Canada is not old enough as a manufacturing country to compete with methods on the other side. We have not in this country as yet the skilled machine hands and workers that the United States has. We cannot get the turnover per machine per man that American manufacturers get. They can turn out materials and parts at less cost per machine per man than we can, although we may have about the same machine and we may have workmen of superior intelligence to some of the men over there. I do not know why this is so; they pay higher wages and produce cheaper.

Some of these conditions can be improved by following along the lines indicated in Mr. Allee's paper. Some we cannot overcome for many years. Prior to the war, manufacturing in Canada was about on a par with production in the State of California. In the last eight years the manufacturers in California have multiplied their production 50 times and are having the same growing pains as we are because manufacturing is something new to them.

Stringent Requirements for Aircraft-Engine Parts

(*Concluded from p.* 784)

workable but also can be constructed at reasonable cost.

EARLE W. PUGHE[3]:—Does the Ex-Cell-O inspection department report to the engineering department or to the production department?

W. F. WISE:—We believe that the inspection department should report to the general manager, so that neither the engineering department nor the factory controls the type of work that is being put through. One designs and the other makes; and we place the inspection department under the general manager, so that there are three distinct executive heads.

MR. PUGHE:—What is the approximate ratio of inspection cost to direct labor cost?

MR. WISE:—About 7 per cent, while it is about 3 per cent for automobile work.

EARL D. OSBORN[4]:—Does that figure include the cost of rejected parts, or to what are they charged?

MR. WISE:—Our rejected parts are charged up to the production department that makes the errors.

[3] M.S.A.E.—Mechanical engineer, Chevrolet Motor Co., St. Louis, Mo.

[4] M.S.A.E.—President, Edo Aircraft Corp., College Point, N. Y.

MR. PUGHE:—Is the production laboratory under the chief inspector, and are the metallurgical tests reported to him?

MR. WISE:—We do not have our own laboratory, for many reasons. We believe that we could show a profit on a laboratory, but we wish to get an unbiased opinion; therefore, being in Detroit, we have all our work done by the Detroit Testing Laboratory. If a factory laboratory finds a carbon content or a vanadium content close to the specification, it may stretch a point, but an outside laboratory like this certifies its analyses and is not at all biased by any direct interest in the results.

CHAIRMAN WARNER:—This discussion indicates the growing appreciation of the rôle of production in the aircraft industry. I believe that a very substantial proportion of airplane factories and accessory factories dealing primarily with the airplane industry still consider production a part of engineering and put the production department under the chief engineer, a practice which certainly is not common among the automobile factories.

Wiring Installations on Pleasure Craft

By C. G. Meeker[1]

Motorboat Meeting Paper

After briefly tracing the historical course of electrical installations for yachts and motorboats, the author analyzes the disadvantages and dangers of some wiring-installation practices that are common and makes definite recommendations for materials and methods that are applicable to craft of various classes. Precautions are outlined for the prevention of injury because of installation conditions and because of moisture, and attention is given to the arrangement of the circuits.

The paper closes with a plea for the formulation of rules for the betterment and standardization of electrical installations.

The discussion touches upon the inadequacy of the electrical equipment in the smaller cruisers and the relation of insurance control to marine electrical equipment.

VIRTUALLY all pleasure boats of any considerable size that were built between 1900 and 1910 were propelled by steam. Most of these were provided with electric lights but with no other electrical equipment. All equipment, such as generating sets, pumps and steering gears, was steam driven. The electric-lighting system comprised a steam or belt-driven generator, a storage-battery for night service, a simple switchboard, and wiring for the necessary number of lights throughout the boat. Most of the larger boats were equipped also with bell or annunciator systems, and a few with telephone systems. The arc searchlight was the only searchlight available.

By about 1910, gasoline engines had been developed to such a point that they were being installed in boats of 100 ft. or longer, and auxiliaries were required to be driven either by small internal-combustion engines or by electricity. As it was impractical to load the boat with small engines, it was necessary to make the electric plants of sufficient capacity to take care of bilge pumps, fire pumps, sanitary pumps and any other power-driven auxiliary equipment that might be necessary.

Electrical auxiliaries were brought into much greater prominence after the World War, with the advent of the Diesel engine for propelling, because of the increased size of the boats and the great amount of power that was necessary to drive the various auxiliaries. The Diesel engine required a large amount of compressed air, as well as fuel-pumps, separators and circulating pumps. The power required to drive these auxiliaries on some of the larger boats was equivalent to 10 per cent of the power of the main engine. Corresponding enlargement of the switchboard, storage-battery equipment and other units was necessary as the power requirements increased. Owners, builders and architects fully realize that the electrical installation on a yacht is next in importance to the main powerplant.

Installation Rules Are Lacking

Unfortunately, from the beginning of electrical installations on yachts, no special rules have existed to govern the method of making these installations. Most specifications have been written without specific details as to the type of equipment or the location of the wiring. This has caused a great amount of confusion between builders and owners, which we think could be entirely eliminated if the same care and attention were given to this part of the equipment as is given to the equipment of other types in a vessel.

When electric lights were first installed in pleasure boats, the same type of wire that was used in private-house work was utilized and covered with wood moldings in almost all cases. Joints were generally soldered; if not, there was no special criticism of the work because there was no inspection. The fire underwriters had no jurisdiction on the water, so any type of electrical contractor could be employed as long as the lights were made to burn after the completion of the job. Ordinary bell-wire was very often used in launches, where the normal voltage was 6 to 12, and installed in a way that was almost sure to cause trouble because little effort was made to make the job watertight or to protect the joints and outlets from moisture.

Iron pipe was universally used until the time of the war in the larger boats that were constructed of steel. At about that time the Navy perfected what is commonly known as Navy cable. This consists of a copper wire covered first with rubber, then with lead, and the whole encased in a woven-steel braid. Because the Navy had adopted it, many architects and builders believed Navy cable to be the most suitable for yacht work. It has become quite common practice to install this type of wire in pleasure boats with no protection at the outlets or junction boxes either to make them watertight or to protect the ends from grounding because of puncturing the insulation by the fine steel braid.

However, this cable does not lend itself readily to pleasure-craft service unless space is provided by the architect or builder for the installation of the fittings that have been developed by the Navy to go with it. Further, steel gutters and ducts are provided when Navy cable is installed on Naval vessels, as the electrical system on board a battleship is its most vital part.

Provisions of this kind are almost never made on a

[1] Smith-Meeker Engineering Corp., New York City.

yacht; the cable is run in any space that is found not occupied by pipes or other construction. The result frequently is that the cable is damaged by nails or screws, causing difficulty after the boat is completed. We have always recommended non-metallic conduit for use back of joiner work where there is not room for iron pipe or outlet boxes, rather than flexible metallic conduit that can be punctured by nails or screws. A nail driven through non-metallic conduit seldom causes any harm unless it actually cuts the wire.

An effort should be made to formulate practical standardized rules for the installation of electrical equipment on private boats. These rules would need to be different in many respects from the rules governing any other class of electrical work afloat, because many things have to be sacrificed to save weight and space in pleasure boats.

Joints Should Be Accessible

We have found that it is common practice to conceal joints behind joiner work and in unaccessible parts of the boat. This practice should be entirely eliminated. Whenever connections are necessary, they should be made inside of metallic junction-boxes or behind fixtures, and no splices of any kind should be allowed where they are not accessible after the boat is completed. Most of the fires due to moisture in a vessel have been caused by concealed splices.

C. G. Meeker

Great care should be taken in the mechanical make-up of a joint, either behind fixtures or in a junction-box. The conductors should be thoroughly cleaned and a good mechanical joint made. This should be thoroughly soldered and taped with both rubber and linen tape, and the finished joint should then be painted with an insulating compound such as P. & B. or asphaltum. This makes the joint moisture-proof and keeps it from corroding.

Two types of wiring that we believe should never be used in pleasure craft, except in exposed work, are the ordinary commercial lead-covered wire, either single or duplex, and the flexible steel-armored cable commonly known as BX. Many boats have been wired with both of these materials, which have probably caused more trouble than any other single class of equipment. If this material is exposed, so that a neat job is necessary, and installed with suitable outlet boxes, it should be satisfactory, but it is not practical to conceal material of either of these types.

Sharp bends are necessary in installing wiring back of joiner work, both to get the wires into the bulkheads and to bring them out at the outlet. Making these bends tends to open the steel or lead casings and is liable to puncture or injure the insulation. Besides, no satisfactory connection can be made at an outlet without using a box which in most cases is too large to be covered by the fixture.

It has been common practice to run wiring of all kinds in the bilge of boats. This should be eliminated entirely, because no type of electric wiring now on the market will withstand salt water for any length of time. As fuel fumes and other gases may be in the bilge, the danger from any short-circuits or grounds that might result from moisture or water would be great.

Wiring Materials for Pleasure Boats

For wiring on yacht tenders over 18 ft. in length, we recommend small brass pipe, brass tubing or brass-armored cable, run along the side of the launch well above the water-line. Several manufacturers of marine fittings make brass outlet-boxes and receptacles not over $1\frac{1}{2}$ x 2 in. in size which can be used in boats of this character, making a very neat, water-tight job. A number of cast-type running-lights also are available, which can be mounted permanently and connections made through water-tight stuffing tubes.

For less expensive boats, such as the smaller tenders and crew's launches, the wiring may be made with super-service all-rubber-insulated twin wire, strapped with brass straps well above the water-line and out of reach of spray. This cable can also be ended in the same type of waterproof boxes that would be used for the more expensive launch. No midget water-tight switch is available. If the standard water-tight switch is considered unsightly, a good quality of toggle or snap-switch can be used if put under the dash at a point where it will not be subjected to spray. These switches are inexpensive and should last two yachting seasons without giving trouble.

If a searchlight and portable cockpit-light are used, these should be on circuits separate from those of the running-lights and protected by fuses. The fuse-boxes and all connections should be well above the bottom of the boat and preferably underneath the forward deck or at a point where they would not be subject to mechanical injury or moisture.

For 40 to 75-ft. boats with wooden hulls, in which 32-volt systems are generally used, we recommend galvanized-iron conduit in the engine compartment and for the feeders running fore and aft. From the end of this conduit, which should stop in an accessible place in a locker or under a seat, the best quality of rubber-covered wire could be used, either protected by molding or run back of joiner work in what is known as circular loom or similar flexible non-metallic conduit.

Arrangement of Circuits

Particular care should be given to the layout of the circuits and location of lights in boats of this size. Generally speaking, separate circuits should be provided for divisions of the boat, as follows: (*a*) owner's quarters; (*b*) deck-lights, if any; (*c*) running-lights; (*d*) searchlight; and as many additional circuits as may be

necessary to provide for the power auxiliaries such as windlass, pumps and air-compressor. Generators on boats of this size usually are gasoline driven and vary in capacity from 600 watts to a maximum of 5 kw. for a 75-ft. boat.

The type of switchboard for the control of such a plant depends entirely upon the amount of equipment in the boat. Every switchboard should be equipped with a high-grade voltmeter and ammeters and a voltmeter switch such that readings from the storage-battery and generator can be taken at any time. All circuits should be controlled by double-pole knife-switches and protected by National Electrical Code cartridge-fuses. The switchboard should be in an accessible place in the engine-room, where it will not be subject to mechanical injury or to moisture entering either through an open port or from the hatches above.

For wood or steel vessels of 75 ft. and longer, still greater care should be given to the location of the wiring. The amount of wiring multiplies very rapidly as the boats increase in size, because of the great number of auxiliaries now regarded by yacht owners to be necessary. In addition to all of the previously mentioned lighting equipment, provision must be made for complete engine-room equipment, including such items as an air-compressor, various fuel-pumps, lubricating-oil pumps and water-pumps, stabilizer and gyro compass. Among the equipment found on the deck are an anchor windlass, a boat hoist and warping winches. It is not unusual to find between 50 and 60 motor-driven auxiliaries on the larger yachts.

Unless a complete system is laid out and the electrical work is put in in a systematic manner, this equipment will take up a great amount of space and difficulty will be found in tracing out trouble or making repairs. The size of wire installed on the larger boats for various loads is important, because many of the hoists and other auxiliaries are subjected to excessive momentary overloads. If the copper provided is of too small capacity, the machines will not respond in cases where emergencies arise.

Preventing Damage from Moisture

On boats of this type, great care should be given to water-tightness wherever wires go through decks or water-tight bulkheads. Any wiring that is exposed to salt water above deck should be installed with non-corrosive conduit, either flexible or rigid. All water-tight receptacles should be kept away from waterways and exposed positions as much as possible. Receptacles that are located on the open deck, such as those for running-lights, gangway-lights and boom-lights, should be turned aft if possible, so that they will not be likely to be filled with water if caps are left off. A small hole should be drilled immediately below the deck in any conduit or fitting that is brought up through the deck, to allow any water that might enter if the receptacle cap is left off to drain out.

The arrangement of the circuits is very important on the larger boats. We have found that the absent and meal-lights on many vessels are connected with the owner's stateroom, or the running-lights on the same circuit with the engine-room. This is a dangerous condition and is very annoying to the owner and officers of the boat. The owner's quarters should always be on entirely separate circuits. The six running-lights of the larger boats should also be entirely separate, as they are if controlled from an automatic telltale in the pilot-house. The deck-lights, including the gangway, boom and absent and meal-lights, should be on one or more separate circuits. Generally speaking, 12 or 15 lights are as many as should ordinarily be put on one circuit.

Much electric heat is now being used in the larger yachts that are provided with sufficient generator capacity. Separate circuits should be run for this service, using wire sufficiently large so that the voltage drop at the heater will be negligible, as the loss in heat is proportional to the square of the voltage drop. We recommend not more than four heaters, or a total of 3000 watts, on one circuit.

As the switchboards on vessels of this type are large, they should be located in an accessible position in the engine-room and far enough from the bulkhead so that the operator can go behind it without difficulty. All of the connections on a board of this type should be made with copper bus large enough to carry the maximum capacity of the generator and storage-battery.

Battery Installations Are Growing

Since the advent of the Diesel yacht, the capacity of the storage-battery has gone up in leaps and bounds. This has been brought about by the fact that all internal-combustion-engine auxiliaries are more or less noisy and many of them cause more vibration than the main engines. These machines are annoying when the vessels are at anchor, and it has been found necessary to install storage-batteries of sufficient capacity to operate the full required load over a number of hours to give satisfactory service to the owner.

Generally speaking, very little attention is paid to the location of the battery when boats are designed. It is essential that care be given to the location of such an important part of the electrical equipment, so that it can be properly serviced and maintained. The most important consideration is that the batteries should be located where they will be well ventilated, both for safety and to prevent annoyance of the owner and his guests by unpleasant fumes.

The cables running from the main switchboard to these large batteries should be of ample size to take care of the maximum demand upon the battery. The demand is likely to be from three to four times the normal discharge rate of the battery, and the wire should be selected accordingly.

The internal communication systems on the newer boats are also very important. Bells, telephone and radio should be installed with the same general care as the lighting system. Nothing but the best quality of rubber-covered wire should be used, whether it is run in armored cable or through conduit. The same care should be given to water-tightness where the various fittings are exposed or where the wires pass through bulkheads or decks. Watertight push-buttons and telephones of a number of different types are available, and they should always be used when such items are required.

The power required for telephone systems should be obtained from a separate source than that used for the bells. The batteries for these systems should be located in a cool, accessible place and not in storerooms or lockers where stores may be piled on top of them.

We believe that it would be to the advantage of architects, builders and owners to get together with the vari-

ous electrical companies in the marine trade for the purpose of formulating a few simple, practical rules for the betterment and standardization of electrical installations on boats of various types. An occasional meeting of the parties concerned would at least tend to clarify the situation and would probably tend to better the general quality of the electrical installations made on new boats.

THE DISCUSSION

H. H. BROWN[2]:—The electric-lighting equipment receives very little consideration in the purchase of a small cruiser. Such a boat usually is equipped with a 6-8-volt generator of the third-brush type. When the purchaser begins to live aboard the boat in summer, he finds it very comfortable but finds that the electric equipment is somewhat inadequate. I have made a substantial improvement by installing a second battery in such a boat. The arrangement that I adopted was such that the battery while being charged was also operating the lamps; but it has seemed to me that a better arrangement might be to use one battery for the lights while another is being charged.

C. CONNELL[3]:—It has been said that the National Board of Fire Underwriters exercises no regulation over electrical installations in motorboats. Has any thought been given to having the Underwriters cover such installations?

CHAIRMAN A. M. WOLF[4]:—My understanding is that the Underwriters have no jurisdiction on floating craft having low-voltage installations of the automobile type; nothing prevents installations such as would not be allowed on land.

L. OCHTMAN, JR.[5]:—Since the Underwriters establish the insurance rates, they certainly have some influence in regard to the installations.

MR. BROWN:—The Underwriters do have indirect jurisdiction, because they can refuse to approve insurance on boats that are not satisfactory. I think no laws in regard to this are necessary, because the owners of boats want to have them insured.

CHAIRMAN WOLF:—While Mr. Meeker's suggestion is fresh in our minds, we should take action to organize a committee for the formulation of standard specifications for the electrical equipment of pleasure boats.

MR. OCHTMAN:—What sort of rubber-covered wire does Mr. Meeker recommend; is it the type known as weather-proof, having a rubber core surrounded by waterproofed fabric?

Mr. Meeker recommends cartridge fuses for the lighting circuits. We consider the screw-type fuse-plug superior.

Author's Reply to Discussion

C. G. MEEKER:—The National Board of Fire Underwriters is maintained by the American fire insurance companies. Marine insurance covers all damage, whether or not it is due to fire, and a large proportion of the marine insurance is carried by foreign companies. It is doubtful whether the foreign companies would be willing to work under the rules of an American association when they have rules of their own which they think apply. The Underwriters do establish insurance rates to a certain extent; but rates vary greatly between American and foreign companies, and American companies refuse many risks that will be taken promptly by foreign companies. This indicates the keen rivalry that exists and the unwillingness to be restricted by definite rules.

In reply to Mr. Ochtman's question, most of the commercial wire is insulated by an inferior quality of rubber. We recommend and use what is known as 30-per cent Para. The approved thickness of rubber for standard No. 14 wire is 3/64 in. We have special wire made up, which we use on boats of 50 ft. or longer, using 3/32 in. of high-grade rubber insulation. The life of this wire is indefinite. We know of one boat that was wired in 1909 and has been almost entirely free from grounds and short-circuits.

The only place for which I mentioned the use of cartridge-type fuses was on switchboards. No switchboard-type switches are available in polished copper with provision for back connection and screw-type fuses. The National Electrical Code requires cartridge fuses for work of this class. We consider cartridge-type fuses more suitable for marine service in general, because receptacles and fittings for screw-type fuses are of cheap quality and much more likely to give trouble on board vessels than are the clips which hold the cartridge-type fuses. The screw-type fuse itself also is of much more flimsy construction and more liable to mechanical breakage than is the cartridge-type fuse.

A. J. SMITH[6]:—In 1929 the Marine Committee of the National Fire Protection Association submitted the attached addition to Appendix D—Internal-Combustion Engines. No objections have since been received, so this section is in line for final approval and insertion at the Annual Meeting of the Association in 1930.

ELECTRICAL EQUIPMENT

Low-voltage installations do not warrant admission of substandard material and workmanship in motor craft where the possible presence of flammable or explosive vapors renders a spark or incandescence from a physical failure liable to entail serious consequences.

(a) Electrical installations operating at potential of over 32 volts shall be in accordance with Appendix B (A.I.E.E. Marine Rules). Those carrying 32 volts or lower shall conform to the following:

(b) Generators, motors and switchboards shall be placed in dry and adequately ventilated locations, as high above the bilges as practicable.

(c) All acid-battery sets should preferably be located in a box or locker on the weather deck; but sets of 16 or less cells may be placed under deck if located so that gas generated in charging can be easily dissipated by natural or induced ventilation. Acid batteries should be set in lead pans.

Battery sets, acid or alkaline, should be secured against shifting with roll of the vessel and should be easily accessible for observation of terminals, testing and replenishment. Preferably, batteries should

(*Concluded on p.* 810)

[2] M.S.A.E.—Red Bank, N. J.

[3] Electric Storage Battery Co., Philadelphia.

[4] M.S.A.E.—Automotive consulting engineer, Newark, N. J.

[5] M.S.A.E.—Chief engineer, Elco Works of the Electric Boat Co., Bayonne, N. J.

[6] Secretary, Marine Committee, National Fire Protection Association, New York City.

Body-Finishing Materials and Processes

By George J. Farnworth[1]

Pennsylvania Section Paper

AFTER a general outline of the painting process, as now practised in automobile body plants, the author describes four modifications of the original pyroxylin finishing methods that have been made because of difficulties with the original all-lacquer methods.

Oil primers are said to be more reliable under cleaning conditions such as are found in practice, particularly with parts in which the last traces of oil are hard to eliminate. Synthetic-resin surfacers introduce economies, because several coats can be baked at once. Improvements in glazing putties and in maroon and light pigments are recorded, and mention is made of new methods of striping and finishing body belts without the need of shielding. Successful experiments with color finishes that combine the functions of the surfacer and finish coats are said to point the way to an ideal color material.

Topics touched upon in the discussion include the feasibility of clear lacquer, the relative durability of different classes of finish, the practicability of synthetic primers and the reasons for difficulties that have been experienced in securing durable finishes for the spokes of wood wheels.

AUTOMOBILE finishes, as applied today, can be divided into three major groups: primer, the function of which is to form a bond between the metal and the succeeding coats; filled or surfacer coats; and the color finishing coats. The first two coats are designated as undercoats.

After the body is cleaned, a coat of primer is sprayed on. If an oil primer is used, it is dried in a steam-heated or hot-air oven at a temperature of 200 to 300 deg. fahr., the drying time varying between ½ hr. and 2½ hr. After this either a knife coat of glazing putty and two or three coats of surfacer are applied or, on bodies made from high-finish sheet, the glazing putty is omitted and one coat each of primer and oil surfacer are applied and baked separately. If synthetic surfacers are used, two or three coats are applied and baked at one time.

Specially prepared sandpaper is used for water-sanding to a fine finish, without pits or scratches, and the body is cleaned and dried before applying the lacquer coats. Two to four double coats of lacquer are applied, depending upon the price of the job; and then, if it is a two-color finish, the portion first finished is shielded with tape paper or a metal shield, and the lacquer is applied to the upper portion of the body. The bodies are then cleaned and dried, and dark colors are given a coat of fairly "long" thinner and light colors are given a solvent coat consisting of about one part of heavy lacquer to three parts of thinner.

Finally, the body is polished, either by hand labor or machine polishers, and touched up where the finish has been rubbed through, then passes inspection and goes to the trim shop. Further touching up is done after trimming, because of mars that occur during trimming.

Modifications of Lacquer Finishing Systems

All-lacquer was the original conception of pyroxylin body finish; that is, pyroxylin primer, pyroxylin surfacer, and pyroxylin-lacquer colored enamels. Now we have other finishing systems which, in the main, owe their conception to the difficulties encountered with the all-lacquer finish. Such systems are:

(1) Oil-base primer, pyroxylin surfacer and pyroxylin color-enamel

(2) Oil-base primer; oil-base surfacer coats, separately baked; and pyroxylin color-enamel

(3) Oil-base primer, so-called one-bake or synthetic surfacers and pyroxylin color-enamel

(4) The one-bake system, consisting of synthetic primer, synthetic surfacer and pyroxylin enamel.

The one advantage of the pyroxlin-lacquer undercoat system is that no large drying-ovens are necessary. Its disadvantages are: (*a*) the high material cost; (*b*) all-lacquer primers have a violent antipathy for oil and grease, necessitating more thorough cleaning than is general in large-production shops; and (*c*) they dry by evaporation only and shrink very appreciably, so that, unless dried thoroughly before sanding down, underlying defects will reappear. These materials, however, are the most economical to use for custom repair or repaint work, in shops where drying ovens are not available.

System (2), using a baked oil-base primer and separately baked oil-base surfacer coats, is probably the most durable now known. Its disadvantage lies in the large outlay necessary for ovens and the time consumed in baking the coats. As these materials dry by oxidation and not by evaporation, shrinkage, if any, is negligible.

The whole fourth floor of our plant in Detroit, approximately 125 x 1000 ft., was formerly used for applying the under coats, which consisted of baked oil-base primer and three separately baked coats of oil surfacer. We have now adopted the new type of material, and this floor is sufficient for the application of all the undercoats and the finishing coats as well. The drying costs have been reduced to about one-third of what they previously were, because of lower temperature as well as less drying time.

Oil Primers Meet Cleaning Conditions

The system that is rapidly coming into use in finishing bodies today utilizes for undercoats a baked oil-base primer and two or more coats of synthetic-resin surfacer. The important function of the prime coat is to furnish an elastic, durable bond between the cleaned metal and the following finish coats, and the most satisfactory prime coat is a baked oil-base primer. This type of primer is the only one of the many developed

[1] Director of laboratory, Edward G. Budd Mfg. Co., Detroit.

which will give consistently good adhesion and elasticity over metal as it is cleaned in production shops today, and it has been developed to a high degree to meet modern methods.

Almost all satisfactory oil-base baking primers are combinations of a very high-grade wood-oil-base varnish and inert pigments, among which iron oxide predominates. Naphtha is used to reduce the primer to the consistency necessary for application. As a general rule, all primers which are satisfactory for use under synthetic or lacquer surfaces are "long" enough to require a baking temperature of 200 deg. fahr. for a period of 1 to 2 hr. These primers have been developed, particularly as to the methods of application, to meet the special requirements of the user. They can be obtained in consistencies suitable to be applied by spraying, to be flowed on, or to be applied by dipping. Body panels were dipped in one Detroit plant last year, but I believe they are being sprayed in the same plant now.

The main disadvantages of both the pyroxylin and baked oil-base surfacers have been overcome by the development of the so-called one-bake or synthetic surfacer. The pigments used in this material are the same as those used in surfacers of the other type, but the varnish base is manufactured from synthetic resins and China wood-oil. It dries partially by evaporation, partially by oxidation and partially by condensation. Two or three heavy coats can be applied in rapid succession and dried at one baking in a temperature and time equal to those for one coat of the old-type oil-base surfacer.

In its working properties, the new surfacer is equal in every respect to the best of the oil-type surfacers heretofore used, it adheres strongly to the primer, it has good filling and good sanding properties, lacquer enamels adhere well to it, and it is resistant enough to withstand the attack of lacquer solvents. It is not as elastic as surfacers of the other type, and oven baking very materially affects ease of sanding. Sanding costs may be doubled by over-baking for ½ hr.

Synthetic-resin primers of the one-bake-system type are not commonly used, as their adhesion is poor in the presence of even slight oil or grease. This diffiulty is especially noticeable in finishing sealed doors in which one part is flanged over the other. The cleaning may seem to be well done, but oil will seep out of the cracks after the doors have gone through the undercoat ovens and the paint will come off within a few months.

New Putties and Pigments

Glazing putties are available in materials of all the types mentioned. Pyroxylin putties are used almost entirely for spot-putty operations after the sanding of the surfacer coats. They dry quickly and give satisfactory filling over small spots. Oil-base and synthetic-resin putties are used to knife-glaze over large surfaces. The latter do not work as readily as oil-base putties, owing to their rapid setting by evaporation, and they must be worked thinner and therefore do not fill as well.

Pyroxylin-lacquer finishing enamels have not materially changed in their make-up during the last two years. They consist chiefly of nitrocellulose, gum, plasticizers, pigments and solvents.

Steps have been made to improve the covering qualities of light colors by the use of titanium oxide as a portion of the white-pigment content, but settling troubles are encountered when a mixture containing more than 50 per cent of this oxide with zinc oxide is used. Another notable development of the last two years is a non-bleeding maroon pigment. Almost any shade can now be used over maroon, but the shade cannot be quite so delicate as it can be when a toner is used.

"Orange peel" or pitting has been reduced somewhat by the use of high-boiling-point solvents in both the lacquers and thinners. This has necessitated low-temperature drying ovens. Sanding and polishing are still necessary to produce the luster and finish required for passenger-car bodies. Spray equipment has not been developed to such a degree that it will eliminate this operation; however, so-called luster lacquers are available which give a finish as sprayed that is suitable for motorcoach and motor-truck bodies. These lacquers are made with a high content of blown castor-oil and harden slowly. Lacquers suitable for application by dipping are available also, and we use them for small parts. They require a certain amount of polishing to give an acceptable finish.

Mechanical Aids to Finishing

Mechanical sanding and polishing machines have been developed to such an extent that they are in universal use in body shops today. The disc type is the one in general use. For sanding and polishing lacquer, the disc is composed of felt or multiple layers of fine muslin. The abrasive medium is a water-base compound using cutting materials similar to the old hand-polishes such as Tripoli, Bentonite, small amounts of emery, and sometimes rouge for color. It is, however, free from waxes, as they ball up on the pad and cause scratching. Some hand work is necessary to take care of details and to remove small circular scratches. The final cleaning is done by using a lambs-wool pad over a felt polishing-head. Experiments are being made with a sanding pad having a central water feed and sandpaper discs, for use in the water-sanding operations on surfacer and lacquer.

Mechanical striping guns have replaced many hand operations in production work. Hand stripers are still necessary, however, for touch-up work. The machine-made stripe is more even and heavier and will last against abrasion longer than the hand-applied stripe. The machines required are yet in the development stage, and are subject to frequent trouble from clogging and failure to cut off quickly, but one that is successful in production is soon to be marketed.

Spray guns that will cut in beads and moldings without the use of tape and paper or metal shields are now being tried out in several plants, with encouraging results. This is a foreign development which will save the industry many thousands of dollars per day if it works out well, as I think it will. Tape-paper cost amounts to about $100 per day in our plant, with a production of 200 bodies per day.

The production body-plant is interested in two things: first, any material or method that will produce a job equal to or better than its present work, for the same or lower cost; second, any material or method that will save time in production, which means money.

Synthetic surfacers, properly handled, have made possible both a better job and a saving in time and money. The same synthetic-resin-base materials are now ground with colors, experimentally, to make color enamels and are being used in production on wood auto-

mobile wheels. A finish with a solid content as high as 75 to 85 per cent is possible with these resins, which would indicate the possibility of some day obtaining the ideal color, a material which will cover solidly, fill, flow out smooth and require no polishing. Test bodies have been finished with such material, using an oil primer, no surfacer, and three coats of the synthetic-resin enamel. Spraying was not smooth enough to eliminate polishing, but a very good job was secured after light sanding and a thin coat of the color.

THE DISCUSSION

J. P. STEWART[2]:—When I consulted a car refinisher about adding a few coats of clear lacquer to the finish of a car, to make it more durable without disturbing the color and striping, he said that such a finish would not stay on. Is that so?

GEORGE F. FARNWORTH:—That can be done. Clear synthetic lacquers are on the market, and the clear lacquer will still stay on, but I think it will not last as long as colored finish. Fenders are being finished by using one coat of colored lacquer and one coat of clear synthetic lacquer, and the results are fairly good. Such lacquer will stay on, but I think it will not last as long as fenders with that finish have been on the road.

J. C. GENIESSE[3]:—Is it better policy for a car owner to leave the original finish for two years and then have the car completely refinished, or should he give it an additional coat at the end of each year?

MR. FARNWORTH:—It is as well to let the finish wear out, so long as it does not rust through. If it is rusting, the rusty spots should be cleaned off and covered up; but if it is merely a matter of the gradual decomposing of the lacquer because of the action of the sun and from polishing, I should wait until the finish had worn through to the undercoats and then have the car refinished.

EARL MYLECRAINE[4]:—What is the best way to remove road tar from a car body?

MR. FARNWORTH:—Many agents are available for use on lacquer. Ford benzol, which is about a 25-per cent solution in gasoline, is effective; also carbon tetrachloride, such as is used in fire extinguishers, and cleaner's naphtha are good for the purpose.

H. P. CLEAVER[5]:—I should like to hear more about the maroon under coats that will not bleed through light colors.

Non-Bleeding Maroon Pigment Excites Interest

MR. FARNWORTH:—We have used in our shop maroon pigments that will not show through at the intersection between colors where a light color has been placed over the maroon. As soon as the film has had time to set, say after 5 min., white lacquer can be sprayed over the maroon and no maroon will show through. We have used maroon from three different makers that will act in this way; and I believe that any lacquer manufacturer can supply maroon of this type if it is specified but it is more expensive.

Question:—Are you troubled with blushing, in the summer time, or with oxidizing?

MR. FARNWORTH:—We find that oxidizing, or bronzing, is invariably due to a maroon that bleeds because of the dye toner contained in it. In the non-bleeding maroon the dye is set in aluminum hydroxide and then thoroughly washed before it is dried and ground. This material does not give quite the same richness of color that can be had with the ordinary toner.

We use a rich toner developed by the General Motors Co., composed of 60 per cent toluol, 30 per cent butylacetate and 10 per cent butyl alcohol. This thinner is used the year round, and we are never bothered with blushing. This trouble is prevalent in some shops where cheap thinners are used.

DALTON RISLEY, JR.[6]:—Why does maroon wipe off more than other colors—blue, for instance—when a car is wiped down in cleaning?

MR. FARNWORTH:—That probably is due to the dye toner in the lacquer, which tends to come to the surface and looks like a bronze coating. Non-bleeding maroons do not break down in that way.

Striping and Classes of Finish

CHAIRMAN WALTER A. GRAF[7]:—Why do the stripes rub off so easily from some bodies, while those on other bodies are more durable?

MR. FARNWORTH:—Hand striping varies according to the way it is applied. Sometimes not enough lacquer is applied, and sometimes the striping lacquer is pigmented so highly that it does not adhere well. A stripe applied with a brush simply sets on top of the other finish, and its durability depends upon whether it is put on thick enough to last. The striping machines which I have mentioned use lacquer that is similar to spraying lacquer only more highly pigmented. The machine applies a large quantity of lacquer, which sinks into the other finish.

Question:—Are full-cold-rolled steel sheets used exclusively for making metal bodies?

MR. FARNWORTH:—All grades are used, from single-pickled to full-cold-rolled sheet. We have used sheets such that about 10 per cent of our bodies have had a finish like the former, and we have painted them by taking them off the line just before the surfacer operation and applying two coats of glaze putty over the bad portions. These bodies then went over the regular line again.

MR. STEWART:—What difference is there in the coatings and in the cost between a cheaply finished body and a very finely finished body?

MR. FARNWORTH:—More surfacer probably is applied to the finely finished body, and more money is spent in rubbing it off. Only two double coats are applied to the cheaper cars, and little time is spent in rubbing. Cars may vary in cost by 200 per cent, but the cost of the material will not vary more than 50 per cent.

MR. STEWART:—Is there a great difference in the life of the two classes of finish?

MR. FARNWORTH:—The life depends upon the film thickness. More finishing material is applied to the more expensive job, and it will last longer than the cheaper job unless too much of the material has been

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[3] M.S.A.E.—Research chemical engineer, Atlantic Refining Co., Philadelphia.

[4] Car Lubricating Station, Philadelphia.

[5] Superintendent, J. G. Brill Co., Philadelphia.

[6] M.S.A.E.—President, Risley, Inc., Philadelphia.

[7] M.S.A.E.—Director of foreign engineering, Edward G. Budd Mfg. Co., Philadelphia.

removed by sandpaper or polish in trying to get a high luster.

J. E. FERNLY[8]:—Can lacquer be applied over chromium plating?

MR. FARNWORTH:—It has been done, but the finish is not very durable. Black finishes, which can be high in their content of plasticizer and gum because they do not need a great amount of pigment, have been applied over chromium plate. The surface is first cleaned in hot alkali solution and wiped dry and again dipped in wood alcohol and wiped dry, warmed in an oven to about 120 deg. fahr., and the lacquer is applied with a brush. It can be applied to small parts, not to large surfaces.

MR. FERNLY:—Could this be applied for refinishing chromium-plated wire wheels that have been attacked by rust?

MR. FARNWORTH:—Rust should be removed before the finish is applied. The work should be dipped in hydrochloric acid for about 2 min., rinsed, neutralized in phosphoric acid, rinsed again; then an oil primer should be applied rather than a straight lacquer. I doubt if this finish would stand up on wire wheels, although black lacquer has stood fairly well on the hub caps of wire wheels.

Question:—What are your routine tests for lacquer, such as wear tests, scratch tests and tests for flexibility?

MR. FARNWORTH:—We have a standard bending fixture for flexibility tests by means of which 3-in. panels are bent over a rod having a radius of ¼ in. Some people use sheet-metal cup-testers for that purpose. This method has the advantage that a long strip or large panel can be painted and tested in many places. For ordinary routine tests, we take cryptometer readings for covering property and viscosity readings and make a flexibility test. From time to time we test shipments of lacquer for rapid breakdown in a weatherometer. Automobile producers expect the finish which they apply to last about two years.

Oil Is Enemy of Synthetic Primer

JOSEPH GESCHELIN[9]:—Are synthetic-resin undercoats now being accepted widely?

MR. FARNWORTH:—Synthetic surfacers are being widely accepted, but not the synthetic primers.

MR. GESCHELIN:—Is there any fundamental difficulty with either the primer or the processes?

MR. FARNWORTH:—Synthetic primers require exceptionally clean surfaces. Any oil driven out around door flanges or joints of front structures will soften a synthetic primer so that it will chip off. Synthetic primers have been used extensively, but those who have used them have returned to oil-base primers. One Detroit manufacturer, who cleans the material by sandblasting, adopted synthetic primers in place of pyroxylin primers.

F. P. SPRUANCE[10]:—I do not know of any one who is using a straight lacquer primer. A one-bake system has recently been installed in a number of the large body-building plants, using du Pont quick-drying primer and surfacer. The primer and glaze coat are being applied directly on the metal, and good results are being secured. I believe their success is largely due to the use of a dry process of cleaning, during which the parts go through an oven with an absorbent material on their surface.

At first the users did not clean the door flanges and had sweating trouble such as Mr. Farnworth has mentioned; but they found that this trouble could be eliminated by spraying the dry-cleaning material on the door flanges as well as on the outside panels, thus producing a surface that is more free from oil than it would be when cleaned by a liquid process. If anything can make the one-bake or lacquer-primer system possible, I think that the dry-cleaning system will.

MR. FARNWORTH:—I believe that synthetic-resin primers will not be used with flanged steel doors, and the industry is coming more and more to steel doors and steel front structures. Virtually all cars outside of the General Motors group have either a steel front or steel doors or both.

Making a Durable Finish on Spokes

MR. STEWART:—Why is it that lacquer finish seems to lack durability on the spokes of wood wheels?

MR. FARNWORTH:—This probably is due to the moisture contained in the spokes, which are not dried thoroughly before painting, or the primer used is not long enough. As a general rule, spokes should be sealed with linseed oil and drier, or something of that sort, before lacquering.

MR. STEWART:—I found trouble of this sort with all of the few cars with which I have had to do; invariably the spokes lost about one-quarter of their lacquer within a few months. Is that trouble general?

MR. FARNWORTH:—If the lacquer flaked off and was not knocked off, the trouble probably was because of poor priming. The trouble is common with the cheaper cars, and the season of the year when the painting is done has some effect. Wheels of many cars are painted in the South and shipped north during damp weather, when they collect much moisture and are not properly dried. We formerly had trouble of that sort on small jobs, but we now dry everything in the oven.

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[8] Foreman, Packard Motor Car Service Station, Philadelphia.

[9] M.S.A.E.—Associate engineering editor, *Automotive Industries*, Philadelphia.

[10] A.S.A.E.—Sales manager, American Chemical Paint Co., Ambler, Pa.

Bodily Steadiness—A Riding-Comfort Index

By F. A. Moss, M.D.[1]

This is the fourth report by Dr. Moss on the investigation of riding comfort at the George Washington University and is a progress report on the measurement of automobile riding-qualities. The previous reports were published in the S.A.E. JOURNAL as follows: September, 1929, p. 298; January, 1930, p. 99; and April, 1930, p. 513. In this report, which was presented at the 1930 Semi-Annual Meeting, the author describes improvements made in two wabblemeters for measuring physiological fatigue caused by riding and the use of two accelerometers to correlate the behavior of the automobile with the physiological results.

Results obtained with two groups of subjects, one consisting of taxicab drivers and the other of university students, are summarized, and the results of preliminary tests of the comparative riding-qualities of different cars as shown by their effects on the subjects are also given.

Two conclusions reached are that the subject's efficiency of performance on the wabblemeter definitely tends to decrease with increase in the length of the driving period, and that a definite, positive correlation seems to exist between the vibrations of the car as recorded on the accelerometer and the amount of fatigue produced as shown by the wabblemeter.

OUR PROGRESS since the last report has consisted in improvements in two of our wabblemeters for measuring fatigue caused by riding in automobiles. Along with the tests for physiological fatigue, we have employed two accelerometers to correlate the behavior of the automobile with the physiological results.

Both the wabblemeters were briefly described and illustrated in the last report[2] to the Society. One of those now being used, designed by Dr. H. C. Dickinson, of the Bureau of Standards, consists of a platform mounted on a joint similar to the universal-joint. Loss of balance is measured in two ways; first, by the number of contacts made when the platform is moved enough to touch the metal projections from the base of the machine at the four corners; second, by the quantity of oil pumped through the machine. At each of the four corners of the machine a piston-like arrangement pumps oil every time the platform is moved, and the oil is collected and measured in a graduated container. Such a method of measuring the wabble enables us to get a measure of the loss of balance, even though it is too small to be recorded in an actual contact or to register on the counter indicating the contacts. This is the machine referred to in this report as Wabblemeter 3, being the third machine constructed in our development of instruments for testing steadiness.

Improvements Made in Wabblemeters

Since our last report this machine has been improved by the installation of an automatic counter, operating by a make-and-break circuit system, which during contact at any one of the corners counts steadily, thus enabling the final contact record to take account not only of the number of contacts but also of the length of time the contact is maintained. Our previous counters recorded only the number of contacts made, giving no difference in record between a contact of 1 sec. and one of 5 sec. or more. This Wabblemeter 3 has also been improved by the adjustment of springs and pump system to measure very small amounts of unsteadiness.

The other wabblemeter (No. 4), as already described[3], is designed to give a graphic record of each wabble or bodily sway. The individual taking the test stands on a platform mounted on a ball-and-socket joint. The movements of this platform as the individual stands on it are recorded by pens registering on a revolving drum timed to make a complete revolution in a given time. One pen records movements from side to side, and another records movements backward and forward. A perfect record, or no movements of the platform, produces a straight line on the chart, and wabbles are recorded as deviations or ups and downs from this straight line. Normal records may be practically straight lines or show some deviations; fatigue records show a larger number of deviations in the record line, in some instances almost continuous deviation first in one direction and then in the other. Fig. 1 shows a normal or before-fatigue record, and Fig. 2 a record taken after a driving test. This type of wabblemeter gives a permanent and detailed graphic record of each test.

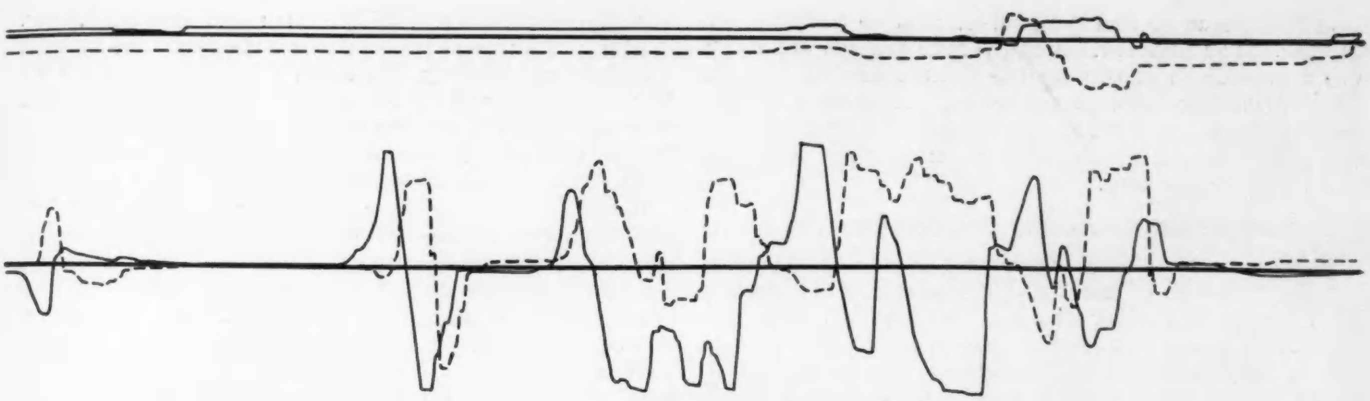
At our last report this machine was in its first trial stage. Since then we have improved it chiefly in the direction of installing a slower motor and a more convenient drum arrangement whereby the record for a 1-min. test can be recorded on a chart approximately 20 in. long. The charted records are converted into numerical scores by using a scaling device whereby each ¼-in. deviation from the midline counts one point for each ¼-in. extent of the deviation. The scoring is done by applying a translucent stencil over the chart (See Fig. 3).

Although improved, both these machines have imper-

[1] Head of department of psychology, George Washington University, City of Washington.

[2] See S.A.E. JOURNAL, April, 1930, p. 516.

[3] See S.A.E. JOURNAL, April, 1930, p. 517.



Wabblemeter Records Taken before and after Riding

Fig. 1 (Above)—Normal, or Before-Fatigue, Record. Fig. 2 (Below) Record Taken after Driving 120 Miles. The Solid Line That Varies from the Straight Base-Line Represents Sidewise Wabbles of the Plate on Which the Subject Stood. The Broken Line Represents Forward and Backward Wabbles

fections, some of which can be overcome fairly easily. Both the wabblemeters are available for demonstration at the Summer Meeting.

In our experimentation with these instruments we have used two groups of subjects: a group of taxicab drivers and a group of University students. Results with each group will be briefly summarized.

Tests of Taxicab Drivers

In cooperation with the Black and White Taxicab Co., of the city of Washington, a group of its drivers were given a series of tests using our Wabblemeter 3. Thirty-nine men and 94 sets of measurements were included in this series of tests, each man receiving from one to eight sets of tests. A test consisted of a wabblemeter record at noon when the drivers were going out and a record at midnight after 12-hr. work of driving, the amount of driving being from 80 to 300 miles but usually between 100 and 200 miles.

Individual Differences.—An interesting factor in the testing of these drivers was the study of individual differences in steadiness of the men, all doing approximately the same work under approximately the same conditions. The distribution of the 39 men in the average number of contacts made before driving is shown

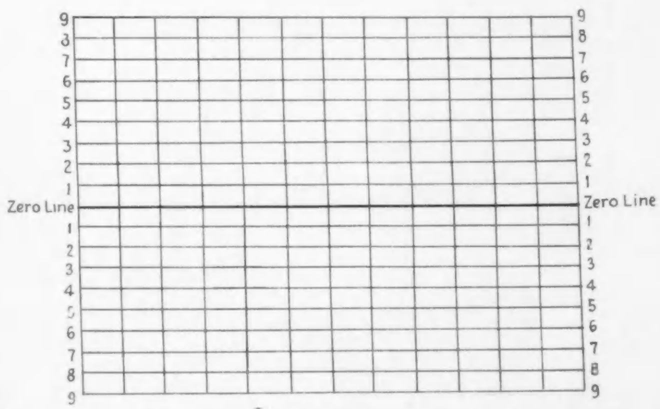


Fig. 3—Translucent Scoring Stencil for Wabblemeter Chart

This Stencil Is Placed over Records like Those in Figs. 1 and 2, with the Zero Line on the Base Line of the Record, and Each ¼-In. Deviation from This Line, as Shown by the Numbered Lines on the Stencil, Counts 1 Point

by the solid line in Fig. 4. The median record is slightly over 35 contacts on the machine in 1 min. The

TABLE 1—RESULTS OF WABBLEMETER TESTS OF TAXICAB DRIVERS BEFORE AND AFTER DRIVING

Name	Miles Driven	Contacts on Wabblemeter Before	After	Oil Pumped through Wabblemeter Before	After
Tests Given Feb. 21, 1930					
Smith	106	18	29	2	4
Proctor	140	24	38	6	9
Henderson	142	35	47	6	8
Meaney	135	42	61	6	8
Crofton	130	45	58	3	5
Emelio	135	24	35	3	5
Farnoff	188	35	45	5	8
Powell	185	21	35	4	6
Murphy	175	35	45	5	8
Stone	155	35	45	6	8
Mangum	140	35	46	4	6
Tests Given Feb. 22, 1930					
Powell	286	50	67	6	7
Chism	175	35	47	4	5
Taylor	190	34	46	4	6
Balman	185	48	60	5	8
Proctor	200	35	50	3	6
Smith	275	17	37	2	4
Clark	246	20	45	3	7
Dunbar	195	34	45	3	5
Slowinsky	220	38	50	3	5
Owens	180	50	65	7	9
Waller	195	24	40	2	4
Bell	240	30	45	3	5
Blankinship	275	35	50	5	7
Tests Given Feb. 24, 1930					
Proctor	130	43	50	5	6
Bradford	125	47	58	6	8
Mayo	115	37	48	4	6
Clark	155	28	41	4	6
Smith	100	39	39	3	5
Benson	110	45	55	4	5
Fielder	108	37	45	6	8
Henderson	145	35	49	6	7
Ritnour	150	26	35	3	5
Fielder	145	28	38	3	4
Gray	148	25	40	2	4
Maxwell	96	37	27	4	4
Falks	115	33	46	2	4
Pator	120	38	50	4	6
Waller	138	38	46	2	4
Bislin	140	31	40	3	5
Costello	130	38	40	4	5
Sanford	152	34	46	3	5
Powell	135	40	48	5	6
Farnoff	140	43	52	6	7

best man made 16 and the poorest man 50 contacts per minute. The distribution of average records after the day's driving is shown by the broken line of Fig. 4. The variability here is somewhat increased and the median record is considerably higher, the median record after driving being 38 per cent higher than the median before driving.

Taxicab and University Groups Compared

It is of interest also to compare these records with the distributions of records for the University group. Both before and after driving, the variability in the University group was greater than in the taxicab group. Before driving, comparable records for 62 measurements on the same wabblemeter ranged from 9 to 50 contacts per minute, with a median of 32; and after continuous driving for a half day or more, from 13 to 90 contacts, with a median of about 46. The median record for this group after driving was 41 per cent higher than the median before.

Normally, or before driving, the taxicab group seems to be slightly less steady than the University group. This may seem somewhat surprising, but certain factors seem to explain the findings. Experience with the machines has led us to believe that age and sex have some influence in this connection. Observations of the wabblemeter records have shown that better records are in general made by younger individuals and by females than by older persons or males. Since the average age of the taxicab group was considerably above that of the University group and consisted entirely of men, the comparative normal records are not surprising.

Practice Effect.—Before progressing very far with our wabblemeter tests, it was desirable to have a check upon the practice effect in performance on the machines. The taxicab group furnished good material for this, since several of the men had repeated tests. A study of successive normal or before-driving records revealed surprisingly little tendency to constancy of improvement with practice. An average of second "normal" trials is practically the same as for first trials. The successive records for six typical subjects who had five or more trials are shown in Fig. 6.

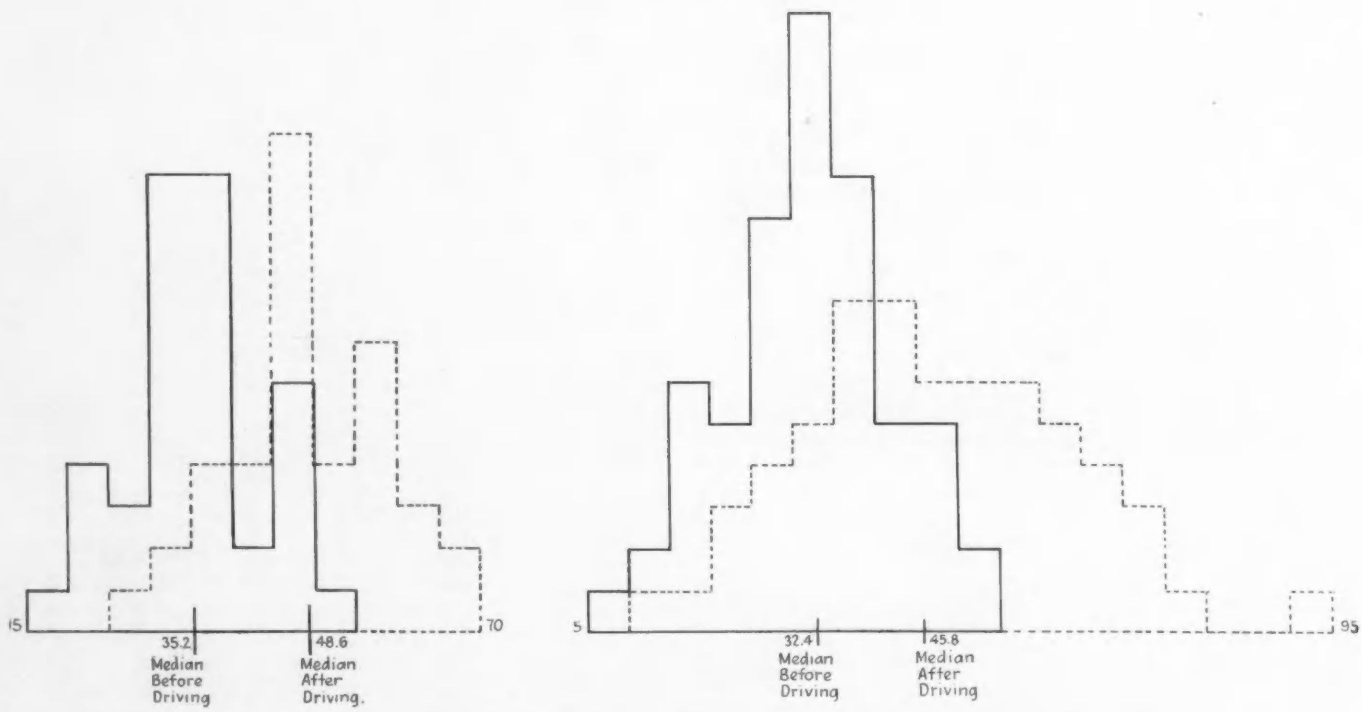
Decrease in Steadiness of Taxicab Drivers After Driving.—For all the 94 sets of tests made, a record was taken of contacts on the wabblemeter and of the amount of oil pumped before going out for the day and after coming in following the day's driving. All the 94 measurements except 2 showed a decrease in steadiness after driving. The causes of the two exceptions are not definitely known, but probably represent the influence of extraneous factors tending to put the subjects in unfit condition even before driving. Typical detailed records for three days' testing are given in Table 1.

Summarizing, the per cent increase in number of contacts and oil pumped after driving are shown in Table 2.

TABLE 2—TESTS OF TAXICAB DRIVERS

Distance Driven, Miles	Increase after Driving, Per Cent In Contacts	Increase after Driving, Per Cent In Oil Pumped
Less than 100	43	44
100 to 149	60	53
150 to 199	56	52
200 or more	69	79

A graphic record is shown in Figs. 7 and 8. Fatigue after a day's driving is definitely indicated by both tests. There is some differentiation according to distance driven, a considerable difference being noticed between the records for the drives of less than 100 and those of over 200 miles.



DISTRIBUTION OF RECORDS FOR CONTACTS ON WABBLEMETER 3

Fig. 4—Taxicab Drivers Fig. 5—University Students

Solid Line Indicates Records before Driving; Broken Line Indicates Records after Driving

Tests of University Students

This part of the investigation consisted in the application of tests on both wabblemeters. Altogether, 36 subjects with more than 150 measurements were used in this part of the experimenting, a large proportion of the same subjects taking road tests under differing conditions of road, distance and so on.

The results of the whole group of tests are summarized in the Table 3 and in Figs. 9 to 12. The tests included in this group were made in the same car, a 1925 model.

TABLE 3—TESTS OF UNIVERSITY STUDENTS

	Increase after Driving, Per Cent			
	Wabblemeter 3		Wabblemeter 4	
Trips	Contacts Made	Oil Pumped	Sideway Wabbles	Back-forward Wabbles
All trips	50	38	37	28
3, 4 and 5 hr., ordinary road	33	42	30	22
All-day trips	57	18	36	29
Half-day, bumpy-road circuit	52	44	43	33

The records of the 3, 4 and 5-hr. trips were obtained from road tests made on the types of roads ordinarily met in long-distance driving, largely hard-surface and improved roads. The all-day trips were made under similar conditions. The last set of figures are based upon results of a number of tests made after driving for 3, 4 or 5-hr. over a very rough and bumpy circuit somewhat more than 2 miles around.

The average increase for all tests involving driving for 3 hr. or more shows 25 to 50 per cent more wabble after driving. It will be noticed from Table 3 and the charts that the greatest increase in wabble is obtained after the tests on the rough-road circuit. The effect on the subjects during the trips might possibly also have been influenced by the monotony of impression, since the whole driving was done around a short, uninteresting circuit.

Comparison of the summarized results for the 3, 4 and 5-hr. trips and the all-day trips shows, on the whole, an increasing fatigue with distance traveled, but there is not a perfect correlation between distance traveled and the amount of fatigue; that is, a day's trip is by no means twice as fatiguing as a half-day's trip.

Comparison of Cars.—These measurements probably will eventually be used in comparing the riding-qualities of different cars, and we have made some preliminary tests along this line. In this part of the testing the same five subjects traveled over the same road on each test, the two cars used being a 1925 model and a current model of the same make of car. Three test trips of this sort showed the results in Table 4. Fig. 13 shows the same data in graphic form.

TABLE 4—COMPARISON OF 1925 AND 1930 MODELS OF SAME MAKE OF CAR

	Increase in Wabblemeter Results, Per Cent			
	Wabblemeter 3		Wabblemeter 4	
Car and Trip	Contacts Made	Oil Pumped	Sideway Wabbles	Back-forward Wabbles
1925 MODEL 120 miles Accelerometer contacts, 39,080	72	15	98	313
1930 MODEL 120 miles Accelerometer contacts, 2,250	48	4	74	103
1930 MODEL 120 miles Accelerometer contacts, 4,360	16	47	311	225

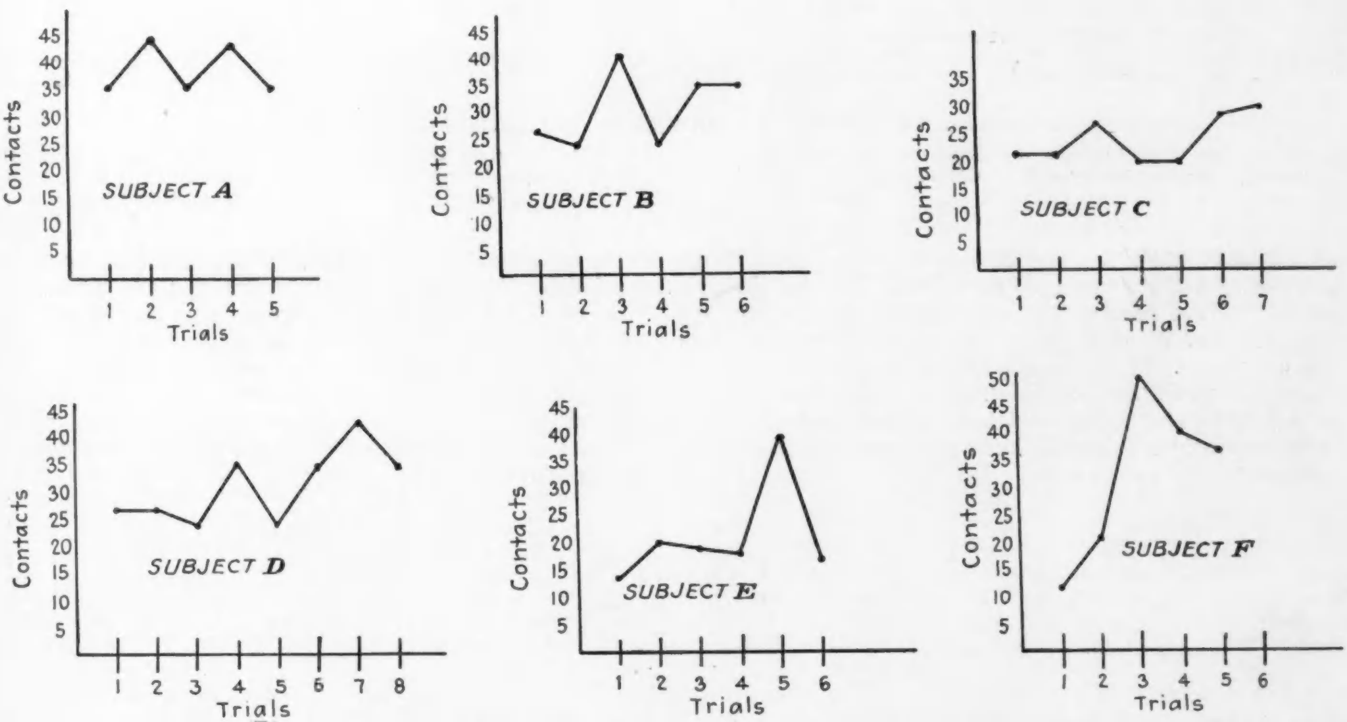
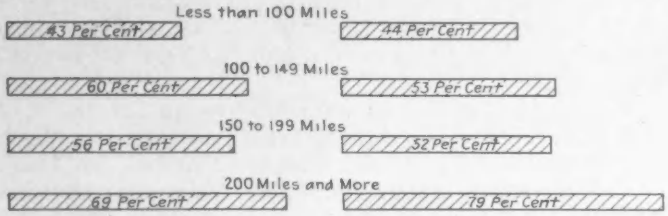


FIG. 6—EFFECT ON RECORDS OF PRACTICE BY THE SUBJECT ON THE WABBLEMETER

Successive Trials, Numbering from Five to Eight, by Six Subjects before Driving. Surprisingly Little Tendency to Constancy of Improvement with Practice on the Wabblemeter Is Shown



INCREASE IN FATIGUE OF TAXICAB DRIVERS AFTER DRIVING, SHOWN BY WABBLEMETER 3

Fig. 7—Per Cent Increase in Contacts

Fig. 8—Per Cent Increase in Oil Pumped

Accelerometer Results.—In these car-comparison tests we have also attempted to get an indication of the difference in the performance of the two cars through accelerometer measurements of the vibrations of the cars. Probably due to our lack of skill in operating the accelerometers, difficulties were at first encountered in getting the accelerometers to register properly, and a considerable amount of data from our early tests had to be discarded. We are now having much more reliable results, and it is planned to check all our subsequent trips with accelerometer readings. Two equivalent trips given show vibrations recorded on the accelerometer for the old 1925 model of 39,080, as compared with 2250 for the newer model. Second trips showed 36,085 for the 1925 model and 2744 for the new model. These results tend to substantiate our findings that the current model is a much better riding car than the old model. It is planned during the summer to continue this work that we can get many more comparisons of accelerometer results with physiological findings.

In conclusion, I should like to call attention to the following:

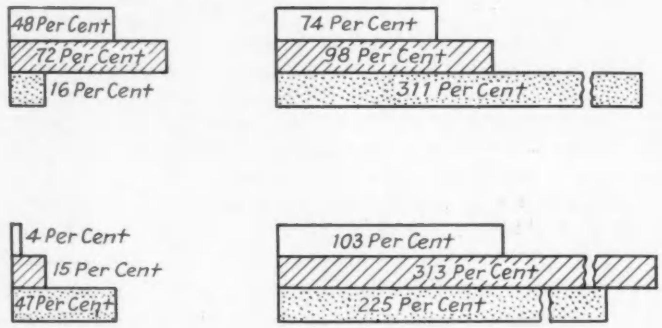
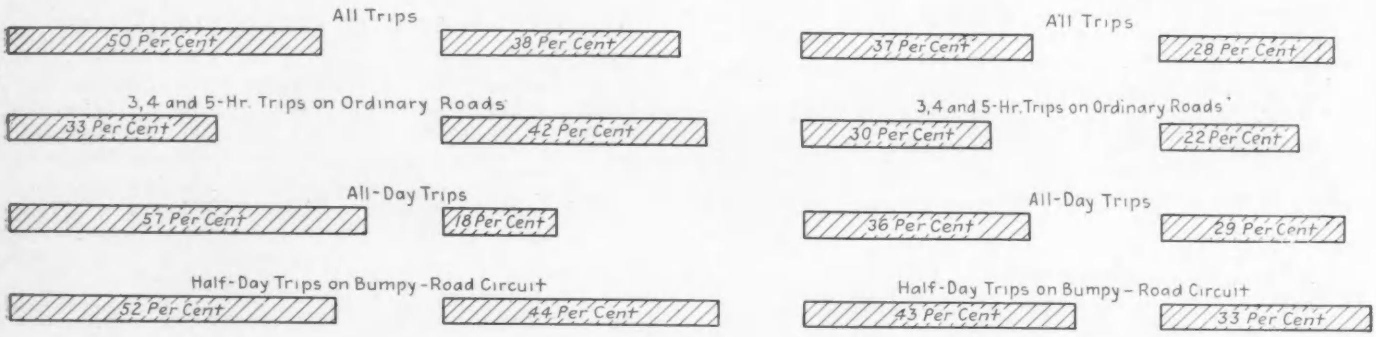


FIG. 13—COMPARISON OF FATIGUE RECORDS OF SUBJECTS AFTER RIDING IN OLD-MODEL AND NEW-MODEL CARS

Percentage Increases After Three Trips by the Same Subjects Over the Same Roads. White Space Represents 120-Mile Ride in Current-Model Car; Shaded Space, 120-Mile Ride in a 1925-Model Car; and Stippled Space, 235-Mile Ride in Current-Model Car

(1) There is a definite tendency for an individual's efficiency of performance on our wabblemeters to fall off with driving. There seems to be some correlation between distance driven and amount of falling off. This was borne out in the majority of the tests, especially in the tests of the taxicab drivers.

(2) There seems to be a definite, positive correlation between the vibrations of the automobile as recorded by the Brown accelerometer and the amount of fatigue or falling off in performance of the subjects on our wabblemeters. This was shown in the tests of different models of the same car, the older model producing both a greater number of registered vibrations on the accelerometer and a greater falling off in physiological performance of the subjects than was found in the new model.



INCREASE IN FATIGUE OF UNIVERSITY STUDENTS AFTER DRIVING

Fig. 9—Per Cent Increase in Contacts on Wabblemeter 3

Fig. 10—Per Cent Increase in Oil Pumped through Wabblemeter 3

Fig. 11—Per Cent Increase in Sidewise Wabbles on Wabblemeter 4

Fig. 12—Per Cent Increase in Forward and Backward Wabbles on Wabblemeter 4

Transportation Engineering

Selling Service-Labor

New England Section Paper Suggests Many Good Service-Station Practices

AUTOMOTIVE service is a basic necessity, said L. T. Hanford, of United Motors, Inc., Detroit, in presenting his paper on Selling Service-Labor before the members and guests of the New England Section on Feb. 13. The mileage covered before necessity for repair or replacement arises has been greatly increased by the improvements in the product, he continued, and automotive service-stations may be heartened by the fact that present-day paved-roads have increased average motor-car mileage within a given time far beyond that known in past years. This results in additional wear that presents a new element to consider in car servicing. People will neglect other seemingly essential personal needs and even appeal for charity at the time they are maintaining some sort of an automobile; but, when their car refuses to start, they will patronize a service station. What one they will patronize depends largely on the equipment, facilities and service expertness of the shop, and how well these features are known to the car owner.

Service economics and sales promotion were the main factors emphasized in Mr. Hanford's address, his first topic being the fundamental economics of automotive service labor. He said in part that the advent of the automobile ushered in an entirely new conception of the term "service" as applied to things mechanical. The very fact that the standard warranty has so long applied only to parts claimed defective, gives recognition to the truth that parts service is the keystone of the arch of all service. Manufacturers therefore found it necessary to make up and stock replacement parts, if the automobile was to become a success. Eventually, the manufacturers were forced to take a greater interest in the repair service of their vehicles. Owners were complaining because their cars were not properly repaired and that the charge for maintenance service was too high, and the dealer or distributor, in spite of his high charges, was losing money in the operation of his service department.

Efficient Service Departments

An efficient service-department usually is a profitable one, and a profitable service-station is usually an efficient one, Mr. Hanford continued. The larger distributors and dealers realized this in connection with their own service departments, and did considerable pioneering research on the subject of "service labor." It is to the everlasting credit of the dealer organization that their efforts were carried out in most cases entirely independently of the manufacturer and, in a few cases, despite the lack of invited cooperation from the manufacturer. Flat rates to owners were evolved and one step had been made in the right direction.

It was about that same time that the matter of better tools and equipment first came into prominence, and some of the early flat-rates to owners took their use into consideration. Better repairs and standardized prices followed. But still the dealer found that he was not earning a fair return on the investment in his service department. Parts prices were fixed and labor charges could not be raised to the customer. This left but one thing to do—decrease the labor cost. Reduction in the wage scale was at best a temporary measure and led to labor troubles, if pursued. The best part of it all was the realization that the productivity of mechanics must be increased in some manner.

Service Labor an Elusive Item

The fly in the ointment is the commodity of service labor. It is a highly perishable and quite elusive item and requires careful research as to its productivity and capable management in its application, Mr. Hanford said. There are too many cases where service managers do not know their final labor costs and their percentage to net labor-sales, he continued. Although service labor cannot be computed on a production basis, it can be computed on a business basis. It is my belief that, in the average service-station in the average city, the labor cost cannot exceed 35 per cent of its sale price. The average working day is 8 hr. and the average month has 25 working days, which equals 200 working hours per month. Actually, to earn a comfortable living wage and return a fair profit to his employer, a service mechanic should perform $500 worth of service labor per month. If he produces $500 worth of labor and he receives 35 per cent of the labor sales, the mechanic should earn somewhere around $175, which, considering a month as four and one-third weeks, would be about $40 per week; with a 48-hr. week, 85 cents per hr., and with a 54-hr. week, 75 cents per hr.

Division of the Service Dollar

A possible division of the service dollar was given by Mr. Hanford, although he said that it varies greatly according to the volume of business, the amount of overhead and other items. The figures are ideal, he said, but they are something to shoot at. The labor cost was 35 cents; non-productive personnel, that is, salaried workers in non-productive work such as clerical help and supervision, 15 cents; rent, 12 cents; light, heat and power, 2 cents; taxes, licenses and insurance, ½ cent; depreciation, 1 cent; shop supplies and materials, 2½ cents; office supplies, 1 cent; telephone, 1 cent; advertising, 4 cents; delivery service, 5 cents; miscellaneous expense, 1 cent; which leaves a total net profit of 20 cents, or 20 per cent. Whatever the compensation plan may be, he believes that these figures are worth remembering, and that the compensation of the men, in its relation to labor sales, is also an important matter that should be remembered.

Inspiration and Incentives Suggested

Something should be done to inspire non-productive workers, said Mr. Hanford. The idea is to make everybody who is working for you a salesman. This plan gives everybody an interest and leads them to become salesmen.

The promotion of service sales, said Mr. Hanford, has been handicapped because in passenger-car service-stations, and perhaps in motor-truck service-stations, a tendency to "let down" has existed because of the fact that it is service for the particular make of car that your company sells and, therefore, there is no competition from other authorized service-stations. But one may have competition from independent service-stations and garages. Do not let down; there are always more customers to be gotten and there is always more to be sold to the customers one has.

"In fact," said Mr. Hanford, "the consensus of opinion among car service-managers is that their best form of sales promotion, and perhaps their most profitable form of sales promo-

tion, is among their present customers. The idea is not to oversell but to sell well. On the average car that is more than a year or two old, anywhere from three to five things ought to be done and, if it were your car, you would have them done. I will venture that if one got into touch with the owners of those cars and brought these things to their attention, he could sell a majority of the various operations to the owner. It represents preventive service, tied up with selling a man a thing he needs."

Some service stations have even tried the plan of paying their service salesmen, men who deserve the name, by commission, and their entire remuneration is on a commission basis on their sales.

Of all forms of service sales-promotion in car and truck service, a letter seems to bring the best results. The other alternatives are personal calls, postcards, telephone calls and the like, Mr. Hanford remarked; but it is alarming to find that only a few service managers know the tangible results of the letters they have sent out. Very few of them know what a given letter costs them, and that is not right! The tangible results of any letter should be known. Good advertising pays, but poor advertising does not; so use the right advertising and keep records of the results, the speaker said.

Many service stations make an offer or suggestion for inspection or something of the sort, and enclose a 2-cent self-addressed reply envelope. Mr. Hanford said he was surprised to find that many service managers did not know of the business reply postcard. Suppose a manager sent 500 letters enclosing 500 2-cent stamped envelopes, which cost 2 cents, without counting the envelope cost. That would be a cost of $10. Say that 40 replies were received; then each letter cost 25 cents. If these 40 people had had a business reply envelope to use, which costs 3 cents, the correspondence would have cost $1.20 instead of $10. The speaker therefore urged the use of the business reply envelope on any occasion in which a reply from the customer is requested.

Mr. Hanford also said that a mailing list is worthless unless it is uptodate. On the follow-up card that asks the customer "Was the work satisfactory? Were our men courteous? Was the car clean?" and the like *the owner's name and address should be typed and typed correctly*. The main reason that more replies are not received is because the owner will not take the time to write his name and address in there. If he could just say "Yes," or "No," or put a check after some question, he would do that. If his name is spelled wrong, he is not likely to reply.

Welcome New-Car Owners

Mr. Hanford advised sending out letters of welcome to new-car owners and said that the owner of a new car is a person whose good will and attendance will repay cultivation. If one cannot make a friend out of a new-car purchaser during the warranty period, he will never be a customer for maintenance. Another point mentioned by Mr. Hanford was to keep the owners' lists—some are kept in a list separate from the service-customer records, and others use the service-customer record-card for the same purpose—carefully corrected. In some service stations different colored record cards for different months of the year are used. In such case, when reference to an owner's card is made, the card shows when the car last came in. The purpose of looking at the file is to decide that, since the owner has not come in for a long time, the manager ought to find out what is the matter. Colored tabs on the cards can be used to avoid rewriting the cards. Use a different color of tab for each month in the year. Then, at the time of year when some work is needed badly and the customers are not coming in fast enough, one can go through the tabs—which are kept in a row according to color—and communicate with the best prospective customers.

Wiring Installations on Pleasure Craft

(Concluded from p. 799)

not be located in the same compartment with a gasoline tank or engine; but where location elsewhere is impracticable, sets shall be effectively screened off by a cage or similar structure in order to minimize danger of accidental spark through dropping of a metal object across terminals.

(*d*) Ignition wiring as supplied by engine makers is generally acceptable. Service wiring shall be not smaller than 14 gage B. & S.

Single conductor shall be National Electrical Code Standard rubber-insulated.

Twin conductor shall be National Electrical Code standard rubber-insulated and lead sheathed.

Armored conductor with substandard insulation is not approved.

Rubber-covered wire may be run in wooden or metal raceways or on fitted insulated blocks where not exposed to oil or mechanical injury, or in conduits or non-metallic flexible tubing in exposed locations such as machinery spaces. However, extended use of conduit is not recommended on account of liability of moisture to accumulate therein.

Wiring joints shall be made up mechanically and electrically sound and soldered and taped.

Battery terminals where soldered lugs are not used shall be frequently examined to ensure proper contacts.

Accessories such as switches, sockets, etc., shall be standard types for current to be carried.

Circuits shall be protected by National Electrical Code fuses of suitable capacity for wire used.

A manual master cut-out switch shall be installed as close to the battery as practicable.

A meeting has been called for Feb. 20, 1930, at which both underwriters and the Association of Engine and Boat Manufacturers will be represented, for the purpose of instituting a marine examination and approval service by the Underwriters Laboratories. This, of course, would include the electrical equipment.

Production Engineering

Hall-Scott Engine Inspection

Chief Inspector Alan Freeborn Tells of Methods Used and Tolerances Allowed

THE inspection department of the Hall-Scott Motor Car Co., said Mr. Freeborn at a recent meeting of the Northern California Section, is answerable only to the factory manager, not to the man in charge of production. This makes the inspectors check and analyze, use or reject all parts passing through their hands in a cold-blooded way.

Inspection begins as soon as the material is received. Castings are examined for general appearance and visible defects. One casting is taken from each lot to the castings-check room, where it is checked against the blueprint to detect any distortion which might have occurred in patterns or core boxes.

As castings pass from one operation to another during machining, the inspectors work in conjunction with the production department. For instance, cylinder-blocks are inspected after the milling, drilling, boring and reaming operations and are checked for leaks and other defects.

Forgings are handled in much the same way. Some forgings which are purchased in heat-treated condition are checked for hardness, the Brinell test being generally used for parts having a low hardness.

Parts that are heat-treated in our factory are inspected before this operation to see that sufficient stock is left so that all tool marks will be removed from the ground surfaces and then inspected after heat-treatment for hardness and flaws. Samples of some parts —connecting-rod bolts, for instance— are tested for tensile strength. The connecting-rod bolts used in our model 160 engine must show a strength of at least 20,000 lb. before breaking.

Gages and Tolerances Used

Every part is completely checked before being assembled into the engine. Besides ordinary snap-gages, we use micrometers and super-micrometers, measuring accurately 0.0001 in., and amplifiers giving direct gage readings of the same accuracy; we have several sets of Johansen gage-blocks which were accurate within 0.000008 in. when checked at the temperature at which they were lapped. For inside dimensions, we use go and no-go plug gages, Bath gages accurate to 0.0001 in., and dial-gage tools reading to 0.0002 in. for cylinder-bores.

Tolerances, sizes and general specifications found by testing and experience to give the best results in finished engines are determined by the engineering department and enforced by the inspection department.

All parts such as crankshafts and camshafts that are rejected are mutilated beyond repair, so that they cannot be salvaged and sold as repair parts. The reason for rejection may be an error of only 0.0001 in., but it is vitally important that interchangeability shall be such that a motorcoach operator can buy and install parts with little or no fitting.

Examples of the tolerances that we hold are as follows: piston-pin diameter, 0.00025 in.; piston diameter, 0.001 in.; piston-pin bore, 0.00025 in.; cylinder-bore, 0.001 in.; crankshaft journals and pins, 0.0005 in.; connecting-rod bore, 0.00025 at the small end and 0.0005 at the large end; and the pitch diameter of the connecting-rod bolts and many studs is held within 0.001 in.

After passing final parts inspection, all work is sent to the stock room, from which it is drawn by the department which builds the sub-assemblies that comprise the complete engine. Inspectors are always working in the sub-assembly department, continually checking such fits as those of piston-pins and piston-rings and general appearance and so forth.

Main crankshaft bearings, which are made of a copper-lead alloy, are fitted by clamping the bearing shells in position in the crankcase and bearing caps in such a way that they are subjected to a crush of from 0.004 to 0.006 in. when the caps are bolted down. With the caps and bearings in position, the crankcase is placed in a fixture and line-bored to size, with a tolerance of plus or minus 0.0005 in.

Engines Run In and Overhauled

During and after final assembly, inspectors check all essential details, including end-play in connecting-rod and main bearings, piston clearances, crankshaft fit, valve-guide fit, run-out of valve seat, and timing.

In the running-in test, each engine is placed on a stand to run for about 4 hr. under its own power at 600 r.p.m., without load, and for 24 hr. at gradually increasing load and speed up to 35 hp. at 1200 r.p.m. At the end of this run, the engine is disassembled and thoroughly washed; the head is checked; the valves are reground; the valve-lifter clearance is reset; and the cylinder-bores, pistons and all working parts are gone over thoroughly, the connecting-rod big-end bearings being refitted and touched up with a scraper if necessary.

Then comes the final test, in which the engine is run enough to warm it up before the cylinder-head is tightened, the valve-lifter clearance set and the ignition checked. During the final run, at 1800 r.p.m., the engine is checked for oil pressure, water temperature and general condition. It is then cleaned and sent to the paint shop.

Taking the Mystery Out of Castings

WORKING of metals is one of the oldest of the arts. At the Production Session of the Annual Meeting, Chairman L. V. Cram said that Noah probably used cast-metal corners in the Ark. If he did, they were made from bronze, because that was the only metal known at the time.

When Noah went to the bronze shop to get his castings, the first thing the foundryman said was: "I shall have to make the patterns myself, because no one else knows how to make them." After adding an extra 1/32 in. or a "scant sixteenth" for draft and putting a big trade-mark on the surface, he turned the pattern over to the journeyman molder, who probably "rapped hell out of it"—not because the casting was not strong enough, but simply to increase the tonnage, because bronze was sold even then on the basis of weight.

Conditions for buying castings did not change materially from the time of Noah to the advent of the automobile. Some time during the last few years, a bright purchasing agent or factory manager decided that castings could better be bought by the piece than by the pound. That somehow has taken a lot of the mystery out of castings, because foundrymen at that time began to realize that foundry practice is an economic problem and not a mystery.

Aeronautic Engineering

Aeronautic Bibliography

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Standardization Progress

Diesel-Engine Standards

Program Adopted by Diesel-Engine Division of Standards Committee at First Meeting

EFFORTS at standardization for Diesel engines has been recognized for several years as one of the important steps to be undertaken in the development and use of this type of powerplant. One or two Diesel-engine representatives have heretofore been appointed on the Engine Division of the S.A.E. Standards Committee to provide the contact for standardizattion as soon as the Diesel interests indicated that it was wanted. With the appointment of the Standards Committee this year, the Diesel-Engine Division was organized, as it was felt that the time has arrived to develop standards that are peculiar to Diesel engines and to extend many of the existing S.A.E. Standards to larger sizes that are suitable for Diesel engine use.

The first meeting of the Diesel-Engine Division was held in Detroit on April 23 at which the procedure of Divisions of the Standards Committee was outlined and a practical program of work for the remainder of this year was discussed. The difficulties that will be encountered in any standardization program for Diesel engines because of the wide range of engine sizes and horsepowers was emphasized and the opinion expressed that probably standardization for the large industrial types of engine will be more limited than for the types and sizes that will be used for automotive purposes.

It was stated, however, that prediction can safely be made that automotive types of Diesel engine will in the future greatly outnumber the larger industrial and marine types and therefore it is important to set up well-chosen standards that can become established in use before Diesel engines are made in large quantities. Although the Division members selected a number of subjects that they felt are appropriate for standardization at this time, they decided to plan the work mainly for the remainder of this year only and to concentrate on a few of the more important and likely subjects.

Subjects Selected for Standardization

The two subjects considered most important at this time are fuel-feed tubing and the connection dimensions for fuel nozzles in the cylinders. In general it was thought that tubing diameters should be standardized over a definite range, with two or possibly three series of wall thicknesses. The Division is to make a general survey of the tubing now used by the several Diesel-engine manufacturers and obtain their recommendations as to a standard.

It was appreciated that standardization of nozzle-mounting dimensions will be difficult because of the great variety now in use, but it was felt that it is very important to the Diesel-engine manufacturers that as they emerge from the present stage of development, the nozzle mountings be guided into as few standardized designs as possible as an economic necessity in the manufacturing and servicing of engines. A questionnaire was outlined for this subject to obtain suggestions from the engine and nozzle manufacturers, with the hope that definite recommended practices can be worked out from the information supplied.

Other subjects listed at the meeting are an extension of the present S.A.E. Standard for Connecting-Rod Bolts up to and including 3-in. diameter, following the S.A.E. or National Fine Series of thread pitches; valves; high-pressure and flared-tubing connections; indicator cocks; fuel-pump mountings and glow-plug sizes. Another of the important items selected is slotted nuts for sizes up to 3 in.

The present standard S.A.E. Engine-Testing Forms are to be reviewed for such modifications as will make them suitable for use for Diesel engines, and a report will be prepared on Diesel-engine nomenclature that, in all probability, will follow somewhat along the lines of the present S.A.E. nomenclature for automobiles.

Division Members in Attendance

Unfortunately, several members of the Division, including Chairman Jahnke, who at the time was in California, were unable to attend this first meeting of the Division. The late L. M. Woolson, of the Packard Motor Car Co., who was one of the first members appointed on this, the first Diesel-Engine Division, met with his fatal accident while the Division was in session in Detroit. Those present were D. W. R. Morgan, vice-chairman, of the Westinghouse Electric & Mfg. Co.; J. F. Fox, of the United States Navy; H. D. Hill, of the Hill Diesel Engine Co.; A. J. Poole, of the Robert Bosch Magneto Co.; E. B. Rawlins, of the Cooper-Bessemer Co.; and R. S. Burnett, manager of the Standards Department of the Society.

Ball-Bearing Standardization

How International Activity Is Leading to Broader Scope of the S.A.E. Work

SINCE the Sectional Committee on Ball-Bearing Standardization was organized by the Society and the American Society of Mechanical Engineers under the procedure of the American Engineering Standards Committee (now the American Standards Association) in 1918, its work has become largely international in scope and has been based very largely on that of the Ball and Roller-Bearings Division of the S.A.E. Standards Committee.

Last January a report on single-row radial ball-bearings of the light, medium and heavy series was approved and is one of the first American Standards of this kind that has resulted from international agreement.

With this standard as the basis, the S.A.E. Ball and Roller-Bearings Division held a meeting in New York City on April 25 to review the present S.A.E. Standards for the separable (open) type, that is also referred to as the magneto type of bearing, and the S.A.E. Standard angular contact type of bearing and to bring them into agreement with the new standard for the single-row radial bearings.

A new project before the Division is to recommend a definite standard for a narrow, light series of annular ball-bearings for use where restricted dimensions and light weight are of greater importance than the load capacity of the bearing. One of the

more important classes of apparatus for which these bearings are used is aircraft engines, and data and suggestions from the aircraft industry were accordingly secured for the Division. A Subdivision report submitted at the Division meeting is being circulated for general approval or constructive comments by the industry before the Division recommends any definite specification for adoption.

Metric Thrust Ball-Bearing Project

One of the projects that has been under consideration for some time for international agreement is metric thrust ball-bearings. This subject was referred to the Ball and Roller-Bearings Division by the Sectional Committee on Ball-Bearings. This type of bearing is used to a very limited extent in the United States as compared with European countries and, because of the appreciable differences in practice in America and abroad, it does not seem to the Division at present that it will be practicable to make all the changes in American practice that would be necessary to meet the proposed standardization of this type of bearing abroad. A Subdivision is, however, preparing a definite report to be submitted and considered at a later date.

A Subdivision report on standardization of adapter-sleeve bearings for both American and English inch-dimension shafts and for metric shaft-sizes has been submitted for comparison with a proposal for an international standard received from abroad through the International Standards Association. These bearings affect primarily lineshaft installations and are not of such direct importance to the automotive industry as other types.

Metric Taper Roller-Bearing Proposal

A subject before the Division and the American Sectional Committee that probably will become important within the next year or two is the standardization of taper roller-bearings on an international basis. A tentative proposal that has been submitted from abroad to the American committees for discussion includes a series of taper roller-bearings made to metric dimensions to interchange with standard ball-bearings; but, as very few bearings of this type are made in America, the Division and the Sectional Committee do not yet know to what extent standards for this type of bearing can be established in the United States. No information has yet been received from abroad regarding standardization of these bearings made to inch dimensions.

Bearings for Electric Motors

A project of rather far-reaching importance to the automotive as well as other mechanical industries in the United States is the standardization of general-purpose electric-motor frame dimensions. One of the related problems is the standardization of antifriction bearings for this class of apparatus. The recommended practice for ball and roller-bearings, approved last January, was formulated largely for use in electric motors and based on the previous S.A.E. Standard for Wide-Type Bearings.

The next step in this program, as requested by the electrical industry, is to establish a definite series of bearing sizes for the series of electric-motor frame sizes and to recommend definite limits for shafts and bearing-housings so that bearings for electric motors shall be completely interchangeable. The Division and Sectional Committee are working on this problem and probably will prepare at least an informal recommendation as soon as a definite report on electric-motor frame dimensions has been finally approved.

Somewhat related to this last program is the proposal to standardize the bearing lock-nuts and washers that are used for many installations. A Subdivision of the Ball and Roller-Bearings Division has been appointed to work out this problem with the several bearing manufacturers.

A canvass made recently among bearing manufacturers in the United States regarding their nomenclature for various types of bearing and the component parts for them indicated some variation in the terms used but not sufficient to make a standard nomenclature impracticable. A Subdivision report on Nomenclature for Ball-Bearings is being circulated and a supplementary report for Roller-Bearing Nomenclature probably will be submitted later this year.

New Aircraft Subjects

An Additional Subdivision Organized To Take Up Fuel-System Parts for Standardization

THE Aircraft Division, meeting on April 9 in Detroit, during the All-American Aircraft Show and the Detroit Aeronautic Meeting of the Society, considered reports of the four subdivisions appointed by Chairman Hardecker at the beginning of the year.

The Reports of the Divisions to the Standards Committee, published as Section 2 of the May issue of the S.A.E. JOURNAL, show the various specifications on pulleys, pulley spacers, streamline and internal tie-rods and flat-head pins which were approved by the Division for submission to the Standards Committee at French Lick on May 25. The result of the Standards Committee's action will be found elsewhere in this issue under the Report of the Standards Committee.

Subdivision 1 under the chairmanship of C. T. Porter, of the Keystone Aircraft Corp., of Bristol, Pa., while as yet reaching no definite specifications, is working on extruded shapes and countersunk aluminum-alloy rivets. It was further decided at this meeting to assign to this Subdivision for consideration all forms of aluminum and duralumin rivets such as the round-head, flat tinners-head and brazier-head rivets.

Subdivision 2, under the chairmanship of Mac Short, of the Stearman Aircraft Corp., has under consideration specifications for streamline tubing shapes and other aircraft tubing. A report from this Subdivision is expected at the time of the next Division meeting, which probably will be held the latter part of August in Chicago during the National Air Races.

The Subdivision on Brakes, Tires, Rims and Wheels is making noteworthy progress, particularly in directing experimental development of semi-balloon landing-wheels and full-balloon tail-wheels. There is considerable interest in the work of this Subdivision and it is anticipated that the results, when completed and adopted, will be of benefit to the entire industry.

A new Subdivision, to be known as Subdivision 5, having the following personnel: Chairman, H. A. Hicks, Stout Metal Airplane Co.; G. C. Emerson, Wright Aeronautical Corp., a representative of the Army Air Corps; and A. L. Parker, Parker Appliance Co., has been appointed by J. F. Hardecker, division chairman, as a result of the discussion held at the Division meeting. The standarization of such fuel-system parts as tube clips, fuel valves, hose nipples, tank flanges, hose and hose liners has been referred to this Subdivision for study and such recommendations as it finds possible to make.

Publications of the Society

(*Continued from p. 718*)

to the members professionally are the papers published, which reveal in accurate detail the most recent design improvements that have found commercial acceptance and research work and experimentation that forecast betterments which may be incorporated in automotive products several years later.

The worldwide prestige of the S.A.E. JOURNAL is the outgrowth primarily of the excellence of the papers presented by men of recognized standing in their professions and which deal with the current problems that confront the engineers. A secondary factor that gives THE JOURNAL its high standing is the extraordinary care that has been exercised from the start by General Manager Clarkson to ensure the technical accuracy, the literary quality and the freedom from typographical errors of the published papers. In the early years of the Society, Mr. Clarkson did all of the editing himself, but as the volume of work grew it became necessary to secure assistants to handle much of this detail work under the direction and supervision of the General Manager, who has consistently required the maintaining of a high standard.

Constant Effort to Improve

The officers, the Council, the Publication Committee and the headquarters staff of the Society are repeatedly and continuously considering ways in which THE JOURNAL and TRANSACTIONS can best serve the needs and desires of the members and will welcome suggestions of ways in which THE JOURNAL can be made even more useful, interesting and attractive, bearing in mind the primary purpose and the restrictions as laid down in the Constitution.

THE JOURNAL is essentially an engineering publication, clearly meant to be of assistance to members in their work, and encroachment upon the fields of the trade periodicals is studiously avoided. It covers fully a field distinctly its own and has a character that those responsible for its guidance believe should be carefully preserved. The activities of the Society have grown so greatly and the interests of the members are so diverse, however, that numerous problems have arisen that have called for much consideration.

One of the major problems is how to provide for the publishing of all the good papers presented at National and Section meetings. In the 12 months ended with April, 1930, a total of 145 papers were presented at 12 National meetings and 202 papers were presented at 128 meetings of the 20 Sections, making a grand total of 347 papers, of which 134, or 38.4 per cent, have been published or are scheduled for publication. In addition, there was extended discussion on most of these papers.

At the Summer Meeting in 1929 the Council authorized increasing the average number of text pages in THE JOURNAL to provide for the publishing of several more papers in each issue. In accordance with this authorization, an average of more than 20 additional pages of text per issue have been published in the eight months ended with April, 1930, as compared with the same period a year ago. This increase accommodated a total of 103 papers with discussion as against 85, an increase of 18 papers occupying 184

S. A. E. Bulletin

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INAUGURATION OF THE S. A. E. BULLETIN

The activities of the Society are becoming so numerous that the Council at its meeting this month authorized the publication periodically of this leaflet as a Society news medium for the members.

STANDARDS COMMITTEE

DIVISION ON WHEEL DIMENSIONS AND FASTENINGS FOR SOLID TIRES.

Conference with Representatives of Tire Manufacturing Companies

Following up the work described at the last meeting of the Society the Wheel Dimensions and Fastenings for Solid Tires Division addressed the following letter to tire manufacturing companies:

1451 Broadway, New York.
March 24, 1911

Re Solid-Tire Wheel Dimensions

In reference to the correspondence which we have had with you on the above subject and the general indication from the tire companies that a conference of their representatives with our Committee is desirable, we would like to learn if you can send a delegate to a meeting at the above office on Friday, April 7th, at 10 o'clock a. m.

The purpose of this meeting is to aid us in determining upon recommendations in the direction of standardization which will bring about the changes desired with due regard to all interests involved.

An early decision in view of the commercial importance of this subject is obviously desirable and in order that you may properly instruct your representative we would submit herewith the result of our consideration on this subject in which we indicate the manner in which this standardization might be effected.

The two principal dimensions to be determined upon are the diameters of wheels to the inside of tire attachment and the width of felloe

With regard to wheel dimensions the 36-in. wheel in common use may be taken as a starting point and larger diameters progressing with increments of two inches. A close examination of the dimensions at present in use by the majority of tire manufacturers indicates that standard diameters might be agreed upon as follows:

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additional pages. The proportion of pages of papers and discussion to pages of meetings news and departmental matter was almost 2 to 1 as compared with slightly more than 1 to 1 a year ago.

Even with the foregoing increase, numerous meritorious papers presented in the season just ended probably cannot be published this year. Each issue now is so voluminous that it is felt that few members have time to read carefully even all of those papers and news reports in which they are directly interested.

Major Considerations Involved

Considerations that the publication department constantly bears in mind and endeavors to achieve are to

(1) Present one or more papers in each issue that will be of direct interest to members in each professional class of the membership

(2) Preserve a balance of papers in the various professional activities classification corresponding to the percentages of members in the various groups

(3) Publish the more important and timely papers as soon as possible after their presentation

(4) Publish discussion with the papers

(5) Print approximately twice as many pages of papers and discussion as of all other matter

(6) Keep the cost of publishing within the appropriation for the purpose

These aims are obviously conflicting and call for constant compromise. All papers presented at an Aeronautic, a Transportation or a Production Meeting cannot be published in the next issue of THE JOURNAL without greatly overbalancing the issue. Nor can discussion of the papers be edited, submitted to the speakers for revision, and published so soon after the meeting. Neither can all good papers given at an Annual or a Semi-Annual Meeting be published in the first issue of THE JOURNAL following the meeting. Numerous good National and Section papers must await publication in the months from June to October, in the closed season for meetings.

Another problem that has long been wrestled with is how to combine the publishing of strictly technical papers and of meetings news accounts in such a way as to preserve the dignity and character of the former yet make the periodical so attractive in appearance and entertaining in text that every member will want to remove it from the envelope and become absorbed in its contents immediately upon receipt. Over-embellishment pictorially and typographically seems undesirable but, on the other hand, forbiddingly dry and heavy appearance and nature is equally to be avoided.

Many improvements have been made in the publication in the last three years with the object of striking a happy medium, and further changes are constantly under consideration.

Sections' Birth and Function

Increase in Number and Membership of S. A. E. Sections Parallels Growth of Automotive Industry

PERHAPS nothing is more indicative of the part that the Society has played and is playing in the lives of individual engineers and in the industry than the growth of local activities throughout the Country.

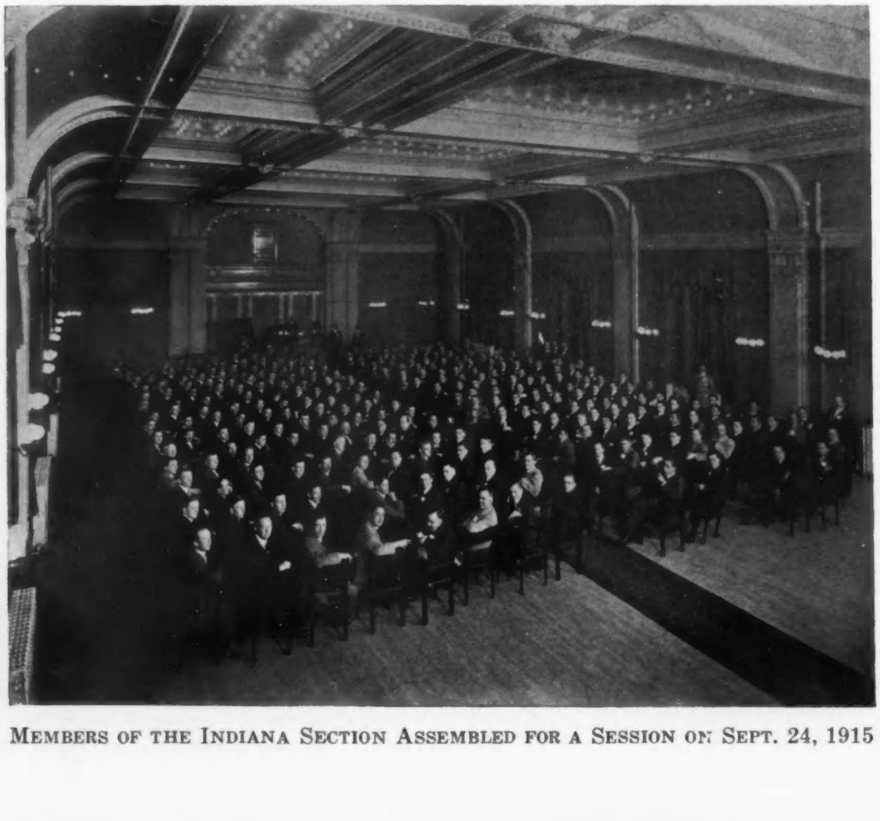
It early became evident that the National meetings alone could not meet the demand which members placed upon them. From this situation arose the organization of local Sections to provide more frequent meetings, an opportunity for discussion of new problems as they arose and an opportunity for sociable contact with fellow engineers in the same locality. The first of such Sections was organized in 1911, when the Metropolitan Section came into being, to be followed closely in the same year by the Philadelphia Section and the Detroit Section. The year 1912 brought the organization of the Indiana Section, and 1914 the Cleveland Section. In this latter year the Philadelphia Section, which had been operating for three years, temporarily suspended its activities.

Notable Years in Sections' History

The following year, 1915, brought the organization of the first Student Branch, founded by a group at Cornell University. The Philadelphia Section was also revived in that year under the name of the Pennsylvania Section, and a new Section known as the Midwest Section, with headquarters in Chicago, started activities. The year 1915 was also notable in Section history for the form of membership then known as the Section Associate Enrollment, which provided that a man might be a member of a local Section and take part in its activities without being a member of the Society.

In 1916 the Buffalo Section was organized; in 1917 the Minneapolis Section; in 1918 a tentatively organized Section was started in the City of Washington; 1920 brought the Boston Section into existence; and in 1921 the Washington Section was reorganized and officially recognized by the Council. In this same year the Dayton Section was started and the name of the Boston Section changed to the New England Section.

Two years later, in 1923, a group of San Francisco engineers started get-together meetings, the first to be held on the West Coast, which culminated in the formation of the Northern California Section a year later. The year 1924 also brought in the Milwaukee Section, and in May of this year the name of the Mid-West Section was changed to Chicago Section. About the same time the Section Associate mem-



MEMBERS OF THE INDIANA SECTION ASSEMBLED FOR A SESSION ON SEPT. 24, 1915

A MEETING OF THE DETROIT SECTION 15 YEARS AGO, IN NOVEMBER, 1915

bership was discontinued and thereafter all members of Sections were required to be members of the Society.

California Sections Recognized

The Minneapolis Section, which started its activities in 1917, was discontinued in 1925 owing to lack of interest and support. A short time later in the same year the Southern California Section came into existence, and both this Section and the Northern California Section were officially recognized, making a total of 13 Sections organized and successfully operating in 14 years.

Denver, Colo., owing to a lack of sufficient S.A.E. members in its territory, has been unable to organize a Section although there is a great deal of interest among the members in that territory in local activities. As a result, the S.A.E. Denver Club was formed in 1927 and, although this organization has never operated strictly as a Section, meetings have been held more or less regularly for the benefit of those interested in automotive subjects. This club is still functioning as such and gives promise of supplying the needs of the members thereabouts until it is found possible to obtain sufficient membership to justify a formally recognized Section.

Aircraft Divisions Formed

During the year 1928 it was found that some of the larger Sections, notably the Detroit, Metropolitan and Southern California Sections, have within their membership individual groups of engineering specialists whose interests fell largely within one activity and whose demands for meetings on their particular subjects had to be met if the Sections were to fully perform their duties.

This was found particularly true in the case of aircraft engineers. The Society has from the inception of this industry been its engineering organization, especially since the merging of the former Society of Automobile Engineers with the American Society of Aeronautical Engineers in 1917. The growth of the aircraft industry brought into the engineering ranks, and consequently into the Society, a large number of aeronautic engineers and resulted in the subsequent growth of the aircraft and aircraft-engine activities in the Society, both in its Sections and nationally.

To provide local activities within the Sections, an Aeronautic Division was organized in each of the three Sections mentioned, with aeronautic engineers as officers and holding meetings designed to suit their particular needs. A similar condition existed in the Detroit Section with reference to body engineers, and a similar group known as the Body Division of the Detroit Section was organized to cover their requirements.

Seven New Sections in 1929 and 1930

The years 1929 and 1930 to date have witnessed the greatest Section growth, seven Sections having been organized in this period. It should be understood that the initiation of any Section activities results, not from any action taken by the Society itself, but rather from a spontaneous demand of members in the various areas to whom the meetings and activities of the Society at short intervals are proving essential.

The year 1929 saw the organization of the Northwest Section, with headquarters in Seattle, Wash., operating over a vast territory including Portland, Ore., 250 miles away. This Section held meetings in each of these cities in order to cater to groups of members who were unable to travel the distance to attend meetings. In the same year the Canadian Section, with headquarters in Toronto; and the Pittsburgh, Wichita and St. Louis Sections were started, bringing the total number to 18. Early in 1930 the Northwest and Canadian Sections were officially recognized by the Council. The other three mentioned as organized the previous year are still operating as probationary Sections and will probably be recognized by the Council some time this year.

It was further found about this time that members of the Northwest Section residing in and about Portland were desirous of organizing their own group and holding meetings more frequently than was possible under their affiliation with the Northwest Section. Consequently the Council gave its approval to the formation of the probationary

TABLE OF SECTIONS MEMBERSHIP

Section	Year Organized	Membership May 1, 1930
Metropolitan	1911	927
Philadelphia (later Pennsylvania)	1911	230
Detroit	1911	1,133
Indiana	1912	216
Cleveland	1914	298
Midwest (later Chicago)	1915	345
Buffalo	1916	141
Minneapolis (discontinued 1925)	1917	
Boston (later New England)	1920	180
Washington	1921	89
Dayton	1921	83
Milwaukee	1924	145
Northern California	1925	125
Southern California	1925	144
Northwest	1929	112
Canadian	1929	74
Pittsburgh	1929	72
Wichita	1929	33
St. Louis	1929	44
Oregon	1930	40
Syracuse	1930	92
Baltimore	1930	47

Section now known as the Oregon Section, which group has been operating very successfully during the last few months and has fully justified the splitting from the original group.

Twentieth and Twenty-first Sections

Likewise, early in 1930, members in and about Minneapolis and St. Paul decided to organize a Section to be known as the Twin-City Section, but, after several preliminary organization meetings, it was found that there was insufficient interest and support and the project was abandoned. In April of this year the 20th and 21st Sections of the Society began their activities. The Syracuse Section, after a preliminary organization meeting, held a very successful dinner meeting at which President Warner presented the initial address. Later in the month the Baltimore members officially started the activities of the Baltimore Section with a very successful meeting held on April 30.

The value of the material presented at Section meetings and the number of members deriving benefits therefrom are impossible to estimate. Each of these 21 Sections will hold, during the coming meetings year, at least one meeting a month for a period of not less than eight months, or a total of 168 Section meetings. This does not take into account the S.A.E. Denver Club nor Section participation in National meetings.

The tabulation given herewith shows the number of members in each of the Sections and the total number of Section members to date. No better indication of the value of local Section activities can be given.

Financial History of the Society

WHEN E. T. Birdsall was elected Secretary and Treasurer of the Society at the first election of officers, in May, 1905, the Society's only financial support was derived from the Annual dues of $10 a year collected from the 52 members. Mr. Birdsall served as Treasurer until 1908, when he was succeeded by Henry Hess.

In this early period the income from membership dues was hardly sufficient to support a paid staff, which was not established until Coker F. Clarkson was appointed Secretary and General Manager in 1909 and four of the founders advanced several thousand dollars to enable the Society to meet its financial obligations.

Under the direction of the General Manager the membership grew sufficiently in the next two years to cover the operating expenses. Of much interest is the fact that the income for the calendar years 1908 and 1909 was reported at $7871.07 and the expenses $6223.56.

In the nine years from 1909 to 1917 the Society had four successive Treasurers. A. H. Whiting served in that capacity from 1909 to 1912, Henry F. Cuntz from the latter year until 1915, A. B. Cumner in 1915, and Herbert Chase for the years 1916 and 1917.

Begins To Acquire a Surplus

Advertising was accepted for the S.A.E. BULLETIN beginning in 1914, to help defray publication expenses, and when THE JOURNAL of the SOCIETY OF AUTOMOTIVE ENGINEERS succeeded the BULLETIN in 1917 the policy of printing advertising was continued. The advertising grew with growth in size and circulation of the monthly periodical and from that year has contributed substantially toward the financial support of the activities of the Society.

Thus, at the close of the fiscal year 1916-1917, the total income of the Society was approximately $122,000 and the organization had a surplus of about $12,000.

C. B. Whittelsey, Sr., was the next Treasurer, and, through his very judicious care of the finances for a period of 12 years, the Society grew in financial strength in proportion to the rapid increase in membership and activities. When he resigned his office in 1929 and was succeeded by C. B. Whittelsey, Jr., the income of the Society was virtually four times the income in 1917, and in the intervening years, as a result of efficient management, such a satisfactory reserve was built up that the organization can well be regarded as on a sound financial basis in this 25th anniversary year of its founding.

Additional Sources of Income

Publication of the standards adopted and recommended practices approved was made self-supporting in 1927. The S.A.E. HANDBOOK was changed in that year from a loose-leaf volume to a bound book issued semi-annually with revisions, and advertising restricted to manufacturers whose products conform to S.A.E. Specifications was admitted for the first time.

Since 1917 the National Automobile Chamber of Commerce, recognizing the value of the Society's work to the motor-vehicle industry, has made annual appropriations to cover in part the expense of the standardization and research work. The American Petroleum Institute has also helped to support the cooperative fuel research.

The following excerpts from the Treasurer's report at the close of the last fiscal year show the very gratifying financial condition of the Society after the first 25 years of its existence:

FISCAL YEAR 1928-1929

Total Income	$405,138.08
Total Expense	386,431.38
Unexpended Income	18,706.70

AS OF SEPT. 30, 1929

Securities	$214,577.00
General Reserve	229,764.86

Aircraft Captivate New York City

Joint Meeting of S.A.E. and the Aeronautical Chamber of Commerce Featured Large Aircraft, Travel and Speed Flying

AVIATION Week in New York City began May 3, with the opening of the Aircraft Salon in Madison Square Garden. The only aeronautic shows that had been held in New York during the last seven years were organized locally and were not sanctioned by the airplane manufacturers' organizations, so most of the airplane companies have been unable to participate in them. The recent show was organized under the auspices of the Aeronautical Chamber of Commerce of America, Inc., and included exhibits from most of the leading airplane manufacturers in the Country.

Recognizing the Metropolitan area as an important center of business and transportation, the Show was suited more to interest the general public than the aeronautic engineer. The transportation facilities of the various airlines and the training and other services of nearby airports were featured. A large public attendance was facilitated by keeping the Show open over two week-ends, the closing date being May 10. Stubs of admission tickets to the Show were redeemable at their cost of 75 cents as part payment on airplane rides at various nearby airports. Extensive Naval aeronautic maneuvers over the city and surrounding country were among the demonstrations that stimulated popular interest.

The engineering end of the subject was well upheld by joint sessions of the Society and the Aeronautical Chamber of Commerce, held in the Park Central Hotel on May 6 and 7. Registration at the technical sessions showed that about 175 different people attended them, and 200 guests enjoyed the Aircraft Banquet on the evening of May 7, held under the same auspices and in the same hotel. The program was arranged and sponsored by the Metropolitan Section of the Society.

Big Transport Planes Massed

One of the main features of the Aircraft Salon was the grouping together on the upper floor of several of the largest airplanes used in transport in America, including a four-engined 32-passenger Fokker, one of the 22-passenger seaplanes of the New York and Buenos Aires line, a Savoia-Marchetti carrying seven passengers in each of two hulls, and other multi-engine planes of well-known makes. Models were shown also of the Goodyear Zeppelin airship that is now being constructed in Akron, Ohio, and of the 100-passenger Do-X and other large Dornier seaplanes.

Small and medium-sized airplanes were massed on the lower floor so thickly that it must have been a puzzle to untangle them when the show was over. The majority of these planes were of the smaller size, with engines having two, four, five and six cylinders. Gliders of several makes also were to be seen, one of them a seaplane glider designed to be towed by a small speedboat. This glider, which has a duralumin hull, is said to take off and fly at a speed of 20 m.p.h. and to land at 10 m.p.h.

Engines and Other Exhibits

Aeronautic engines of nearly all the best-known makes were exhibited. Aside from the best-known radial engines, there were air-cooled, vertical and inverted engines. Among the less conventional designs were a two-cylinder opposed engine; a radial engine having six cylinders equally spaced but with alternate cylinders offset from each other and working on a double-throw crankshaft; and an 18-cylinder W-type Italian water-cooled engine said to weigh 1350 lb. and to develop 1650 to 1800 hp. at 2700 to 2850 r.p.m. The Packard Diesel engine also was there. One of the features of the Packard exhibit was a demonstration of how difficult it is to burn the fuel oil used in the engine when it is not atomized. It was even shown that the oil could be used to put out a small flame burning in a saucer.

Model airplanes, materials, accessories and fuels were among the exhibits that were distributed around the walls. One of the oil companies showed models representing many different forms of transportation, beginning with the Indian drag and including the covered wagon, Fulton's steamboat, two of the first American locomotives, the famous Selden automobile and the

GENERAL VIEW OF A SECTION OF NEW YORK AIRCRAFT SALON IN MADISON SQUARE GARDEN, TAKEN ON SECOND FLOOR

Kitty Hawk airplane of the Wright brothers. These exhibits were made with suitable settings and on various scales, some of the larger animals having motion effects.

Transportation features rather than design features were emphasized at this show, which was primarily intended to sell air transportation to the metropolis. Many agencies cooperated to make the show a success. The United States Navy contributed a force of approximately 150 airplanes, which flew in formation over the city on two occasions, during a trip that took them over Southern New England;; the local police had one of their amphibians on exhibit in Times Square; and one of the newspapers exhibited an immense collection of aeronautic pictures at the show, while all the local papers gave much space to the various events.

Subjects at Technical Sessions

The joint technical sessions were held Tuesday and Wednesday morning and afternoon, May 6 and 7. The Tuesday morning session was presided over by Past-President George W. Dunham, and papers on engine subjects, written by George J. Mead, of the Pratt & Whitney Aircraft Co., and W. F. Davis, of the Fairfield Engine Corp., were presented. Marine Air-Transport was the subject of the Tuesday afternoon meeting, at which the design and uses of amphibians were expounded by Guiseppe Bellanca, while Jerome C. Hunsaker presented the advantages of the airship. Edward V. Rickenbacker, the former racing driver who became the first American ace, was Chairman of the Wednesday morning session, at which Lieut. C. H. Schildhauer read a paper by Dr. Dornier on the great Do-X flyingboat and W. Laurence LePage dissipated some of the fog that has surrounded the Autogiro aircraft, as it is being developed by the Kellett Aircraft Corp.

The Wednesday afternoon session was presided over by Theodore P. Wright, of the Curtiss Aeroplane & Motor Co. Former Lieut. A. J. Williams gave some of his experiences and observations in regard to speed flying at this session, and Major Leslie MacDill presented a paper on Commercial Aviation reflecting the point of view of an officer of the United States Air Corps.

The various technical sessions were held in the ball room of the Park Central Hotel; but this was not large enough for the Aircraft Banquet, which was held in the Colonial Room, the mural decorations of which are quite suitable for a meeting of this sort. They show scenes from the early history of Manhattan Island and include representations of many modern forms of air transportation. Henry S. Breckenridge, a director of the Transcontinental Air Transport, displayed a wealth of wit as toastmaster; and former Commander E. E. Wilson proved a most enjoyable speaker as he substituted for the Chamber of Commerce official who had been scheduled as the chief speaker.

Types of Airplane Engine Discussed

In-Line Liquid-Cooled versus Air-Cooled, and In-Line versus Radial Engines Compared

"IT IS a natural course of evolution that the Society should be the guiding spirit in aircraft development," said Chairman George W. Dunham, consulting engineer, "just as it has been a leader with regard to passenger-cars, motor-trucks, motorcoaches, motorboats, farm tractors and other development along automotive lines." The occasion was the opening of the New York Aeronautic Meeting by an Engine Session May 6, and nearly 100 members and guests were present. A further accomplishment of the Society, according to Chairman Dunham, has been by way of coordinating engineering interests, interchanging valuable data and the standardization of parts processes.

Before introducing the first speaker, Chairman Dunham requested that someone move to extend to Coker F. Clarkson, Secretary and General Manager of the Society, an expression of regard and sympathy because of his ill health for some months past. On motion of Herbert Chase, duly seconded, the assemblage unanimously extended an expression of regard to Mr. Clarkson and its sincere hope that he will speedily recover and return to his appointed work in the Society. Two papers were presented and were followed by constructive discussion.

Comparisons Made on Engine Cooling

George J. Mead's paper on In-Line Liquid-Cooled versus Air-Cooled Engines was read by T. E. Tillinghast, of the Pratt & Whitney Aircraft Co., of which Mr. Mead is vice-president. The paper dealt mainly with conditions more or less generally applicable to commercial aviation. After reviewing the general development of aviation powerplants and analyzing the reasons behind the changes in engine types as a guide to what may develop for the future, the author went into some detail as to the powerplant requirements of commercial aviation of today, mentioning dependability and minimum weight per horsepower as the most important. Speed is one of the principal advantages of aviation, and he said also that we can expect strenuous competition between the various transcontinental lines in this regard, which will not only force close scrutiny of the way in which the available power is used, but will direct attention to the development of larger and larger power-units.

Among other commercial requirements, Mr. Mead said that passenger comfort needs to be improved, which will necessitate quieting the propellers by reducing their speed with gearing. The engine exhaust will need to be muffled and conducted well aft of the passengers, and engine or propeller vibration will not be tolerated. In his opinion, engine units of greater power will have a greater number of cylinders than at present, so that vibration will be reduced to the minimum. Costs will always be important as regards powerplant fuel, maintenance and depreciation.

Assuming that larger power-units are apt to be favored, Mr. Mead said that experience seems to show that 600 hp. is the maximum power which can be applied efficiently to one crankpin. He said further that the inference is that larger units can be developed which have several crankpins, but that air-cooling seems ideal only for cylinders disposed in a single plane and possibly for cylinders disposed in two planes, one of which is behind the other. He doubts that cylinders radially disposed in more than two planes can be cooled satisfactorily.

In conclusion, Mr. Mead stated that it seems that we cannot predict with any certainty whether the liquid-cooled in-line engine or the air-cooled type will be favored for the larger power-units. From the history of aviation, it is safe to say that whichever type becomes standard in the larger powers, the weight of the powerplant per horsepower will be comparable with that of the air-cooled radial unless a gain in performance is secured because of reduced engine-drag sufficient to compensate for its additional weight. In aviation, we cannot afford to take any chances, he said; consequently, the standardization of new types is bound to take considerable time both for thorough development and service testing.

Means of Decreasing Weight per Horsepower

In the discussion following Mr. Mead's paper, Messrs. R. W. A. Brewer, Sanford A. Moss, Chairman Dunham and others made general remarks on

means for decreasing weight per horsepower of aircraft engines. Herbert Chase gave a brief description of the Junkers company six-cylinder engine, which is a two-stroke opposed-piston engine operating on the Diesel-engine principle.

It was remarked by Mr. Brewer that engines can be run very much hotter than had been thought posible and mentioned that a recent development which includes the use of ethyl glycol has made it possible to run liquid-cooled engines much hotter than formerly. He said cooling is a matter of intimate contact between the cooling medium and the part to be cooled, and that liquid cooling provides possibilities of greater cooling from the standpoint of better contact.

C. H. Biddlecombe, of New York University, New York City, remarked that insufficient emphasis has been placed upon reliability. In his opinion, the reliability factor is the one factor that seems to him practically absent in the liquid-cooled engine. A. C. Hewitt remarked that compression ratios are being increased at present to secure better fuel economy faster than the oil companies can supply better fuels, and that this is true also of the liquid-cooled engine. In his opinion, "we are right on the tail of the oil refiner and, while liquid cooling offers this chance of decreasing the hot-spots, we must guard against going ahead without considering the limitation in present-day commercially available fuels." Mr. Tillinghast said that there is much truth in Mr. Biddlecombe's belief that possibly, in considering high-temperature cooling, we are looking toward some type of engine which may have less reliability. In his opinion, under some conditions of bad detonation, an air-cooled engine might cause trouble while a liquid-cooled engine might be able to carry on under such conditions.

In-Line versus Radial Aircraft-Engines

The second paper was presented by W. F. Davis, chief engineer of the Fairchild Engine Corp. Mr. Davis said in part that it is fortunate that the 1929 depression in the aircraft business turned the attention of the industry from the engineer's point of view to a serious consideration of the pros and cons of various types of aircraft engine. He said also that engineers must approach such consideration fully aware of what experience has taught us both as to what has and has not been accomplished, but that they must not be deluded by statistics or figures derived from a period of the industry's history that clearly was dominated by commercial considerations and not by engineering research or development. The present condition of the industry clearly shows, he said, that much remains to be done in aviation research and aviation engineering before the industry will assume a rightly important position.

Mr. Davis stated that his paper was limited to a discussion of various features that bear on the selection of in-line and radial-type gasoline-engines, and discussed reliability as the first consideration. He said that the inverted in-line air-cooled engine, due to the ease with which it and the entire valve-gear can be full-pressure lubricated, and to the simplicity and rigidity of its block type of construction, presents features which cannot be duplicated in the radial-type engine. The rigid-block-type construction of an in-line engine with overhead or underhead camshaft valve-gear practically precludes the possibility of a cylinder-head or cylinder-stud failure, since the loads of each cylinder are transmitted to all the others through the cam-box casting, and the factor of safety is greatly increased. On a radial engine, each of the cylinders stands alone, and cylinder-head and stud failures are all too common.

GEORGE J. MEAD
Vice-President of the Pratt & Whitney Aircraft Co., Whose Paper on In-Line Liquid-Cooled versus Air-Cooled Engines Was Read by T. E. Tillinghast

Simplicity of design leads to increased reliability, Mr. Davis remarked. This involves both the simplicity of design of individual parts and the number of parts in the engine. The ease of manifolding an in-line engine for good distribution and volumetric efficiency permit the elimination of direct drive or geared-up blowers which have been found necessary on many radials to give satisfactory performance. The geared supercharger on an in-line engine is necessary only for special very high performance or for duplication of sea-level performance at altitude.

The radial engine can be made lighter in weight for a given cylinder-displacement, Mr. Davis continued, due largely to the duplication of functions of the crankcase and crankshaft parts. However, the restrictions placed upon its permissible maximum speed due to the load of the heavy rotating connecting-rod assembly and the high inertia-loads of the push-rod-type valve-gear, make it easily possible for the in-line engine to equal or better the specific weight per horsepower by increased output at a high speed with no sacrifice of reliability.

Improved visibility is, in Mr. Davis' opinion, an obvious feature of the inverted in-line engine, and reduction of head resistance is also another point in its favor. When the inverted in-line type of engine is installed on the wing, the advantage it has is that about 80 per cent of its projected frontal area is below the propeller center-line so that the natural flow of air to the upper surface of the wing is undisturbed.

From the viewpoint of the consumer, Mr. Davis stated, the ultimate cost of an aircraft engine consists of its first cost, its installation cost and its service cost. The type of engine has a bearing on all these costs. Unfortunately, there are no figures available for direct comparison from actual records. Such a comparison must be made for several sizes of engines of each type, of equal quality, and based on actual manufacturing costs in like quantities and on actual service records over a period of time.

Therefore, Mr. Davis merely expressed his opinion on the foregoing subject. He said that the cost of tooling for a radial engine is somewhat less than for an in-line engine. In limited quantities, standard machine-tools are somewhat more suited to radial-parts machining than to in-line engine-parts. Hence, in limited quantities the manufacturing cost of a radial engine should be somewhat lower. In quantity production, however, these factors become insignificant and the specialized high-production machinery which has been developed for the automotive industry and which is readily adaptable to in-line aircraft-engine production should offset this difference in manufacturing cost.

Comparisons Between Radial In-Line Engines Made

During the discussion Glenn D. Angle said that he had selected at random a group of radial and in-line engines. All are representative types of commercial engines. In the list there are four five-cylinder engines, one six-cylinder and one seven-cylinder, among the radials. Among the in-line types, there are four four-cylinder and two six-cylinder engines. In comparing

these on the basis of weight per horsepower, he found that in comparing the four and the five-cylinder engines, which of course gives a distinct advantage to the in-line engine because it has one less cylinder, the radial engine has a greater displacement per pound of weight by 10.6 per cent. Comparing a six-cylinder in-line and a six-cylinder radial, the advantage was 21.7 per cent; and comparing the six-cylinder in-line with the seven-cylinder radial, the advantage is with the radial by 21.2 per cent.

Mr. Angle said also that the average of all of these engines shows that the radial engine has an advantage of 17.3 per cent. He stated his belief that 17.3 per cent is a fair average because none of these engines was selected on the basis of low weight. He said further that the cowling of a radial engine would have only very little of the area not measured in the figures he stated which would be covered up; but that with the in-line engine, the cowling would go entirely over it and the effect would be increased by several per cent. Therefore, he continued, the figures probably would bring the comparison to a point where the in-line engine would not show such an advantage as automotive engineers have been led to believe.

In reply, Mr. Davis said that his company has made certain comparisons between certain sized radial engines of equal horsepower with that of in-line engines and, in every instance, the frontal area or resistance, both projected and parasitic, has been found to be much less and in favor of the in-line engine.

Mr. Davis mentioned that his company has built six in-line engines. Previously, he built air-cooled engines of various types. His point was that he and his company have done practical work on the problem. In regard to the cooling, he said that the in-line engine undoubtedly requires less air for cooling purposes. The air surrounds the cylinder barrel almost completely in an in-line air-cooled engine which, in most radial-engine installations does not occur unless the air is forced by baffling around the cylinder.

Various points brought up in addition to those mentioned included statements that there has been too much separation of thought and idea between the aircraft designer and the engine designer and that greater cooperation is needed, the lubrication of valve gears, and that present conditions justify the placement of great emphasis on the subject of improved appearance and greater attractiveness of airplanes so that they may make a stronger appeal to the flying public.

When Doctors Disagree

Relative Merits of Amphibians, Airships and Landplanes for Oversea Service Debated

DIFFERENCES of opinion among aeronautic engineers developed at the Marine Air-Transport session on Tuesday afternoon and made it so interesting for the 100 in attendance that they temporarily forgot the unbearable heat in the ground-floor room of the hotel as well as on the street.

C. H. Biddlecombe, of New York University, presided as Chairman in the unavoidable absence of Virginius E. Clark, who was slated for the job. Mr. Biddlecombe, formerly operations manager of the Colonial Air Transport, also apologized for the inability of Giuseppe Bellanca, author of the paper on Amphibian Design and Transportation, to be present to present the paper in person because of pressure of business in connection with the sale of six airplanes to the Canadian Government. The paper was read by Richard M. Mock, consulting aeronautic engineer.

The only other paper, on Transoceanic Air-Travel, was presented by Jerome C. Hunsaker, vice-president of the Goodyear Zeppelin Corp., formerly air attaché in London and Commander in the Air Service of the United States Navy. But in the discussion Colonel Fitzmaurice, who flew across the Atlantic in the airplane Bremen that landed at Greenley Island, in Canada, gave an analysis of the subject of transoceanic flying that was equivalent to a good prepared paper and won hearty applause.

Problem of the Amphibians

In the advocacy of amphibians, Mr. Bellanca asserted that the public has been awaiting and demanding for a long time the successful development of a plane that can travel on land, on water and in the air, and that can take off vertically and land equally well in a small place. Although we do not yet know how to satisfy all these demands in one craft, the demand for an amphibian is being satisfied. He recalled that Langley's and Bleriot's early flights were made over the water, Bleriot's first three aircraft being fitted with pontoons.

The greatest difficulty to be overcome in the designing of amphibians is sufficient reduction of weight and air resistance, according to Mr. Bellanca. He gave weights of the Bellanca Pacemaker medium-size cabin airplane which has a pay-load of 1042 lb., and said that the addition of floats reduces the pay-load by 500 lb., while the further addition of landing-gear would decrease the pay-load to 292 lb. At the same time, these additions decrease the speed from 145 to 135 m.p.h.

The same plane can be licensed as a seaplane with a pay-load of 1042 lb. If landing wheels are added, they will decrease the pay-load to about 800 lb., but Mr. Bellanca believes a way will be found to increase this to 1000 lb. Landing-gear weighs from 5 to 7 per cent of the gross weight of a plane, or roughly about 30 per cent of the pay-load; and, if exposed, reduces the speed approximately 6 per cent.

Wheels Retractable in Floats

After commenting on the appearance of successful amphibians after the World War in England and America, and expressing admiration for the low resistance, speed and maneuverability of the Loening amphibians, Mr. Bellanca commented upon the difficulty of combining the landplane and seaplane in a machine that will perform as well as either in its own element. Retractable wheels present one of the real problems. In one of his experimental planes the wheels were retracted into the lower stub wings to make a landplane easily convertible into an amphibian by adding two pontoons.

Recently an interesting new type of amphibian made its appearance. This is the Aeromarine Klemm, which is a monoplane on floats, each float having a well at the front step in which a wheel slides up and down, projecting below the pontoon when the plane is on land. Mr. Bellanca pronounced this system quite intelligent.

At present the Bellanca company is experimenting with a small single-engine plane, substantially like its Model-K sesquiplane, which is intended to be convertible from a landplane to a seaplane or amphibian. The use of pontoons with landing wheels inside of them promises to be very successful, resulting in light weight and low resistance. Tunnel tests run some time ago showed that the open well on the under side of the pontoon created considerable resistance, and Mr. Bellanca therefore thinks that a way should be found to close the well.

In conclusion, the author mentioned the advantage afforded by the amphibian of landing safely either on land or on the water.

Amphibian Use Declared Limited

Use of the amphibian in this Country is more limited than appears at first thought, asserted A. A. Gassner, of the Fokker Aircraft Corp., in opening discussion on the subject. In most of the Country north of the meridian of New York City the lakes and rivers

are frozen from December to March, making use of the amphibian out of the question. And landing on small rivers and ponds, especially in a side wind, is as unsafe as a forced landing anywhere. The higher price of the amphibian is another objection, although he believes this type will be very useful to the private owner and will come into extended use in the near future. His company believes, however, in converting the flying-boat into a usable amphibian, because the flying-boat is regarded as more seaworthy than the seaplane. Mr. Mock said that sight should not be lost of the fact that Savoia has had great success with the twin-hull flying-boat, which is quite as seaworthy a craft as the twin-float seaplane.

Weight and Cost Depend on Designer

Regarding weight of the amphibian, W. R. Herfurth, of the Ireland Aircraft Corp., remarked that the additional weight is not that of the landing wheels but of the hull or pontoons. The Ireland five-passenger biplane flying-boat requires a 425-hp. engine, whereas the seaplane of the same size fitted with pontoons performs equally well with a 300-hp. engine. Chairman Biddlecombe added that the main consideration is the sturdiness of structure and stability in rough water that can be secured with a hull as against the relatively flimsy pontoon construction. S. C. Finger, consulting aeronautic engineer, said that he cannot see much future for the twin-float flying-boat, as the float gives as much trouble as the pontoons give. The weight of either the flying-boat or the amphibian is dependent upon the ingenuity of the designer, as is also the cost; the cost of amphibians that are giving very satisfactory service ranges from $7,300 to $17,500.

Attention was called by George Page, of the Curtiss Aeroplane & Motor Co., to the fact that the Italian Savoia-Marchetti Co. builds a single-hull flying-boat that is identical in gross weight with the twin-hull types with which we are familiar in this Country, and that it carries more useful load and has a higher all-around aerodynamic efficiency. Necessary protection against corrosion is a factor that accounts in part for the relative expensiveness of the amphibian as compared with the cost of land airplanes.

Mr. Mock referred to two new types of amphibian that have not yet been commercially developed. One is the single-wheel type developed by Grover Loening, which is a flying-boat with small wing-floats; the other was developed in Detroit and has a hull with close-coupled wing-floats in which the landing wheels are housed, thus combining the stability and seaworthiness of the flying-boat with good aerodynamic efficiency.

Over-Ocean Safety of Airships

Drawing an invidious comparison between airplanes and airships, Mr. Hunsaker, in his paper, pointed out that the records show that three airships—the British R-34, the Los Angeles and the Graf Zeppelin—have made four trip for refueling and for increased safety. He showed lantern slides of these routes, mooring masts and the huge Akron hangar for airships, designs of the new Goodyear-Zeppelin airships and of the Graf Zeppelin and so forth.

PRINCIPAL FIGURES AT MARINE AIR-TRANSPORT SESSION

Capt. C. H. Biddlecombe (Upper Left), of the Aviation Corp., Chairman; Giuseppe Bellanca (Upper Right), Author of the Paper on Amphibian Design and Transportation; J. C. Hunsaker (Lower Left), Vice-President of the Goodyear-Zeppelin Corp., Who Gave the Paper on Transoceanic Air-Travel; F. C. Mock (Lower Right), of the Bendix Aviation Corp., Who Read Mr. Bellanca's Paper

westward and four eastward crossings of the Atlantic and the Graf crossed the Pacific in three days, all successfully; whereas the numerous disastrous transoceanic airplane attempts have shown the hazards of trying to cross the Atlantic or the Pacific in either landplanes or seaplanes.

The speaker dwelt at length upon the reasons for the safety and success of the airship flights and for the danger and failure of the airplane flights. He discussed the several air routes over the Atlantic and the Pacific Oceans, the weather conditions likely to be encountered and the available landfalls offering possibilities of breaking the

Airships are primarily suited for long-distance service and airplanes can render coastwise service, according to Mr. Hunsaker, who holds that the two are non-competitive also because of their relative size and cost.

Fitzmaurice States His Views

Colonel Fitzmaurice said that Mr. Hunsaker's paper had given him a very broad appreciation of the situation with regard to lighter-than-air craft. He has always favored the heavier-than-air craft but believes that we must not consider the two as separate entities; all means of transportation are inextricably interlocked and interdepen-

dent. The only advantage aircraft possess is speed. As compared with the airplane, the speed of the airship is too low, the initial cost is very high, dock and mooring-mast cost is high, and operation and maintenance of the ships and ground facilities are terrifically expensive. He said he thinks there certainly is a future for airship service from London to India and Australia and also over the Pacific Ocean, but that the airplane will eventually carry passengers, mail and some valuable freight over the Atlantic.

Weather conditions over the North Atlantic are admittedly bad, but, said Colonel Fitzmaurice, "if we are going to sit down now and say that after 10 years of air-transportation development we will not get airplanes, engines and ground organizations at least 500 per cent more efficient than we have today, we should stop the development of heavier-than-air craft altogether." He believes it is quite possible today to operate a mail service between New York City and London with heavier-than-air craft.

Lack of Capital a Handicap

Inability to interest capital in large amounts is delaying this development. Germany is the only country in which the government has given the airplane manufacturer financial support. The Junkers 338 four-motored plane with engines inside the wings has proved to be a wonderful success, said the speaker, and it is only a step to the Junkers G-1000, of which he has seen the plans. All that is necessary is the money to build it, and this will be forthcoming. He believes this machine will be in the air within three years and will be a tremendous step forward in development.

Refuting a statement made by Mr. Hunsaker that lack of visibility is the greatest handicap to transoceanic airplane service, Colonel Fitzmaurice stated that blind flying is one of the simplest things in the world. Eventually it will be much easier to land an airplane in fog than to bring a big liner into a fog-bound harbor. If seadromes can be safely anchored they will greatly help to advance the development of heavier-than-air traffic across the Atlantic.

Harold H. Brown, in further discussion, thought that the choice between the two types of aircraft lies in the relative cost of airplanes and airships, seadromes and hangars, and of operating and maintenance costs. Mr. Hunsaker placed the cost of a 100-passenger airship at $4,000,000 and of dock facilities and mooring masts at about $2,000,000 at each terminus. These would serve four ships. Colonel Fitzmaurice said he thought that the cost of airplanes to carry 100 passengers the necessary distance would be much less than that of the airship, and that the seadromes would cost much less than the airship hangars.

Chairman Biddlecombe closed the discussion with the remark that we should consider the millions of dollars already spent for more than 1000 airports in this country that have cost from $200,000 to $4,000,000 each.

Do-X and Autogiro Expounded

Design and Building of the Big Flying-Boat and Principles of the Autogiro Are Recounted

EDWARD V. RICKENBACKER, vice-president of the Fokker Aircraft Corp. and the first American ace in the World War, acted as chairman of the Wednesday morning session of the joint aeronautic meeting. In introducing him, Mr. Rickenbacker said that Lieut. C. H. Schildhauer's 17 years' experience as a big-ship pilot qualified him for the honor that has been granted him to command the Dornier Do-X in its flight across the Atlantic during the coming summer. He spoke of the flying-ship as one of the most interesting experiments in heavier-than-air craft that has ever been undertaken, and considered the most surprising thing about it to be that its sponsors had the courage to appropriate the funds necessary to undertake its building.

Lieutenant Schildhauer presented the translation of Dr. Dornier's paper on the Do-X flying-ship, which was originally presented by Dr. Dornier before the Wissenschaftliche Gesellschaft für Luftfahrt. This was illustrated during the reading by the original lantern slides. Dr. Dornier's paper was published in the May issue of the JOURNAL, beginning on page 554.

Movie Shown of Do-X in Flight

After presenting the paper, Lieutenant Schildhauer showed a motion-picture film of the trial of the big seaplane over Lake Constance. This showed the plane loading, taking off and flying, with the lake, woodlands and cultivated fields in the background. The Dornier plant in Friedrichshafen was shown clearly in the film, and the Zeppelin works in the distance were pointed out by the speaker.

Lieutenant Schildhauer said that, while the Do-X is designed for 100 passengers, it can be used as a troop transport for distances of about 500 miles carrying 160 fully equipped troops. As a bombing plane, it will carry eighteen 1100-lb. bombs for the same distance; and it has emplacements for eleven machine-guns and two large-caliber guns. There is no blind spot on the whole ship. He also said that Dr. Dornier is projecting a 9000-hp. machine having a wing span of about 229 ft., a length of about 180 ft. and a gross weight of between 180,000 and 185,000 lb.

After the Wright Conquerer engines are installed, Dr. Dornier contemplates extended flights to the Mediterranean and other points. If these flights prove satisfactory, the flight to America will be undertaken during late July or early August, at which time the speaker said that the most favorable weather conditions are found over the Atlantic.

Chairman Rickenbacker voiced the sentiment of the meeting, which was approved by enthusiastic applause, in wishing well to Lieutenant Schildhauer in his transatlantic flight and trusting that he will not be called upon to undergo a second initiation to the Caterpillar Club.

Autogiro Blades Have Much Freedom

In introducing W. Laurence LePage, of the Kellett Aircraft Corp., Chairman Rickenbacker recounted his service on the aeronautical research staff of the British Air Ministry and at the Massachusetts Institute of Technology, where one of his investigations had to do with the aerodynamic design of racing-yacht sails.

Without attempting to give a complete mathematical analysis of the forces and principles involved in the Autogiro, Mr. LePage presented a paper which began with a brief history of the development of the autogiro and made its operation relatively clear for a mechanism that differs so radically from that of any other aircraft. He said that the idea of a lifting system comprising a freely rotating windmill was not original with Juan de la Cierva, but that he was the first to make a comprehensive engineering study of it and to face the problem of a symmetry of lift resulting from the difference in air-speed of the two sides of the windmill during forward motion of the craft.

A basic feature of Señor de la Cierva's development is the free mounting of the windmill blades on horizontal hinges at the hub. When rotating, the blades are subject to the action of centrifugal force and the upward lift of the air through which they are passing. This results in their assuming continually shifting positions, pointing somewhat upward and at different angles on the two sides of the hub as viewed from the front. The mast upon which the hub is mounted

generally is set slightly to one side, to make up for the unequal lift at the two sides, and the lift that is transmitted from the blades and the hubs is the upward component of the pull in which centrifugal force is an important factor.

Centrifugal Force Reaches Big Figures

The centrifugal force on each blade of a 2000-lb. Autogiro is said to be about 5000 lb. at 150 r.p.m., so that the inertia forces are approximately 10 times the lift forces. In the latest models, the blades are allowed a restricted motion in the path of their rotation, controlled by interblade bracing cables having shock-absorbers, to relieve certain unbalanced forces that occur periodically because of fluctuations during the cycle.

While the machine has a maximum angle of incidence for maximum lift, an angle of attack beyond this does not result in a stall. The climbing speed is not claimed to be as good as for an equivalent airplane, but the angle of climb is said to be greatly superior. Recent progress made in the development of the type includes an improved means for starting the rotor system in motion on the ground, so that the cumbersome double-surface tail-plane is no longer required.

In answer to questions, Mr. LePage said that vertical descent in the Autogiro is possible from any altitude, but it is possible to see much better and control is much better if the landing is made with some forward motion. The stability is said to be very good, but little flying has so far been done under conditions of poor visibility.

Research, Experiment, Manufacture

U. S. Navy and Air-Corps Officers Contribute Papers on Speed Flying and Commercial Aviation

THE session devoted to research, experiment and manufacture began at 2.30 p.m., Wednesday, May 7, with Theodore P. Wright, chief engineer, Curtiss Aeroplane & Motor Co., in the chair. Lieut. A. J. Williams, Jr., U.S.N., contributed the paper on Airplane Speed-Flying, and the subject of Commercial Aviation in the United States, from the viewpoint of an Air Corps Officer, was presented by Major Leslie MacDill of the United States Air Corps. About 100 members and guests were present.

Speedy Flying Described

In his address, Lieutenant Williams told his hearers in part that when we were capable only of traveling 100 m.p.h. in an airplane, a speed of 150 m.p.h. seemed impossible to attain. When we reached a speed of 175 m.p.h., we thought it marvelous. Then we went up to 200 m.p.h. and heard from all sides expressions to the effect that this was the maximum and about as far as we could carry out air-speed development; but, in 1923, we reached a speed of 250 m.p.h. Then the speed jumped to 280 m.p.h. on a closed course, and from that to 328 m.p.h. around a closed course. These facts are stated to show that we do not know what the possibilities ahead of us are.

The business of flying is merely reckoning power against drag, Lieutenant Williams continued. As we lighten our engines, increase our horsepower, cut down the drag, and continue to concentrate on one objective, speed, none can tell where we will wind up.

The most important thing about concentrating on high-speed research is, said Lieutenant Williams, that it puts within the grasp of the engineer, the promoter, and others who have to do with aviation, an ambition that has been realized by someone else. Therefore what small part in air-speed development the individual may be taking seems but natural to him. Young men of 16 years of age today will be aviators some day. Their psychological reactions are entirely different from his own and others who started to fly during the war.

Our business is to develop the air leg of the airplane journey, Lieutenant Williams remarked. It is difficult to sell airplane rides between two given points if flying at 100 m.p.h. If a 30-mile headwind is encountered, that is a sad condition, and when we compare

CHAIRMAN AND SPEAKERS AT THE AIRPLANE DESIGN SESSION

Edward V. Rickenbacker (Top), Vice-President of the Fokker Aircraft Corp., Chairman; Lieut. Clarence H. Schildhauer (Lower Left), of the Dornier Corp. of America, Who Presented the Paper by Dr. Claude Dornier on the Do-X Flying Ship; W. Lawrence LePage (Lower Right), of the Kellett Aircraft Corp., Who Read a Paper on the New Autogiro

it with the excellent train service in a Country such as ours, where we compete with the train service, we have got to step that air leg up considerably to justify our existence and pay dividends. No feature in this high-speed development is more important than the collaboration between the engineer and the pilot. It would be possible to construct an airplane aerodynamically which would give promise of exceedingly high performance; but, like any efficient engineering project, it always runs through the compromise stage. One of the important compromises we must take care of relates to the needs and necessities of the operating personnel. There seems to be a perfect willingness and a ready means of collaboration between engineers, even of various countries. We are collaborating on engineering data, but very little is being done in collaborating on operating experience. That is further from standardization than is the more technical side.

"When people ask: 'Why do you young people want to be flying around in a pursuit airplane at a speed of 250 m.p.h.?'" said Lieutenant Williams, "I say that I may want to get to some place in a hurry, I may want to get to somebody in a hurry, or I may want to get away from somebody in a hurry. In an aerial combat, one necessity is to get there with alacrity, another is to do your business when you get there, and the third is to get away if the situation should become unhealthy. If one can go through life and create a little bit of healthy dissatisfaction, I think that person is leading a useful life. But it is hard to strike a medium and to placate all individuals without disrupting current thought."

Lieutenant Williams discoursed on the background of the grouping of the various instruments on the board, described vividly some of his personal experiences in racing planes, and mentioned some of the difficulties of landing at say 80 m.p.h. with comparatively flat-bottom pontoons on sea-surfaces that looked perfectly calm, but in reality were rough. However, he said, the hollow deep pontoons used in the first Mercury race removed all difficulty of holding the craft perfectly straight.

Some Aspects of Commercial Aviation

Major MacDill confined his opinions largely to the actions and reactions of commercial and military aviation, considering military aviation as embracing all the aviation activities which have to do with the defense of this Country in war. An accurate estimate of the size of commercial aviation today is difficult to obtain, he stated. The rapid growth of the investment in commercial aviation is interesting to the military person when he attempts to estimate the rate of growth in this branch of aviation and, judging from the past, what the effects of this growth are on military aviation. The expansion has been so rapid, he said, that any one knowing the difficulties that arise with expanding organizations concludes at once that the growth cannot continue at the recent geometrical rate unless the earning power is such as will warrant the tremendous expense and waste of so rapid an expansion.

"Before one can discuss the effect of commercial aviation on military aviation," Major MacDill remarked, "the purpose of military aviation and the way it will be used must be considered." In his opinion, the tremendous size and extent of the World War made people forget preceding wars and he thinks there is an all too prevalent impression that the next war in which we may unhappily be engaged will necessarily be a war of the nature designated by military persons as a "war of maximum effort." By a war of maximum effort is meant a war which in its opening phases is conducted between two powers of such equally balanced military strength that the war is prolonged to a stage permitting each nation to organize all its resources in an effort to win.

It is the desire of every nation contemplating a possibility of war that its military preparedness will be so complete that a condition of war will not exist, said the speaker. It is the hope of each that the will to fight in the opposing nation will be broken at an early date, and that the war will not

PRINCIPAL PARTICIPANTS AT THE SESSION ON RESEARCH, EXPERIMENT AND MANUFACTURE

(Left to Right) Major Leslie MacDill, of the United States Air Corps, Who Made an Address on Commercial Aviation; Theodore P. Wright, Chief Engineer of the Curtiss Aeroplane & Motor Co., Chairman; Lieut. A. J. Williams, Jr., U. S. N., Who Presented a Paper on Speed Flying

be prolonged but will instead be won by superior preparedness in the opening engagements. A country such as ours, with its tremendous resources so well distributed geographically, does not contemplate that its will to fight will be destroyed at any time before the resources of the whole Nation have been completely assembled, and we give great consideration to the formulation of war plans involving the assembly of all our resources.

Two Classes of War to Be Considered

The military personnel must always consider two classes of war, said Major MacDill. One is the war of maximum initial shock and the other is the war of maximum final effort. The war of maximum final effort, and the aviation industrial needs of such a war, have been so thoroughly discussed in the years since the World War that he hopes the effect of the growth of commercial aviation on our preparedness for a war of initial shock will not seem unduly emphasized.

Continuing, Major MacDill said that supremacy in air-power represents a superiority in knowledge of the fundamental facts of aviation, in the technical skill with which this knowledge is turned into equipment, and in the organization and training of personnel using the equipment. Technical skill without superior knowledge is handicapped in the preparation of superior equipment. The best organization and training are handicapped without superior equipment. Consequently, the military person continually desires to be assured that in the exchange of knowledge between ourselves and other nations we will receive the equivalent of what is given, and that in the exchange of equipment we are not permitting our advances in one line to be copied without at the same time receiving something from which we can copy the superior advances of another nation in another line.

Commercial aviation is not necessarily interested in the equalities of such exchanges, but is primarily concerned with present and future earning power, the speaker said. The development of commercial aviation in many ways aids the development of military aviation. One of the greatest assets which military aviation may have is that contributing to an ability to concentrate in a limited space of time an air force placed in various sections of this tremendous Country of ours.

After discoursing on the possibilities of converting commercial airplanes into military airplanes and such subjects as bombardment planes immediately converted from transport planes, training airplanes, and the parallelisms in the use of equipment on the part of commercial and of military organizations such as in the use of engines, Major MacDill said in conclusion that he thinks we are approaching a period of consolidation which will eliminate the great duplication of effort that has characterized the past, and that real technical advances will be made so that aviation can progress to the point at which it is self-supporting.

Following the presentation of the papers, remarks were made by Roland Chilton, O. F. Allen, Ralph H. Upson, Kenneth S. Cullom and others regarding the points which had interested them in the papers. Among other questions, Mr. Chilton asked Major MacDill for suggestions of the manner in which the Air Corps can cooperate with the manufacturer to overcome the delay of perhaps years before a company that fosters a military engine of increased horsepower can expect to cash in on it, and mentioned the reluctance of the bankers to sponsor such a program. Mr. Allen spoke with regard to the exchange of technical data and of the coordination of engineering effort between given organizations both local and foreign. Mr. Upson mentioned that the military spirit is primarily one of competition, and that this must be so because it seeks not to help but to destroy the other fellow. Fortunately, he continued, in America and in general industry, a more cooperative spirit seems to be continually making its way toward the idea of helping each other rather than that of fighting each other. Mr. Cullom voiced his opinion that today a multi-engine airplane has a real advantage. The principal idea is to have a factor of safety which will prevent disaster in case one engine goes out of service.

In conclusion, Major MacDill said that he admitted his guilt in urging people to build large engines, and that he attempted to argue it out from both standpoints. If someone believed in the single engine, he wanted to show that a single-engine school of thought would lead to the necessity for building a larger engine. But that if one believed in three-engined airplanes, he wanted to emphasize that this school of thought also should lead to the necessity for building larger engines.

Major MacDill's paper contains many basic principles of military strategy and of commercial aviation, said Capt. Walter C. Thee, of the U. S. Army, in presenting his discussion of the paper. He said also that the United States enjoys a potential superiority over other nations in that it is the most highly industrialized nation at present. Commercial aviation combined with our industrial superiority is assisting materially to give our Country the supremacy of the air which is necessary to obtain decisive victory in case of emergency.

Captain Thee believes that information pertaining to commercial aviation —in the form of knowledge, technical skill, equipment and organization— should be released freely to other nations because such procedure constitutes part of the code of ethics in engineering and other provisions to interchange both knowledge and ideas. For this reason, he said, conventions and world engineering-congresses are held. In his opinion, it would not be practical to prevent the sale of equipment abroad and, even if this could be prevented, commercial developments could not be kept so secret as to prevent developments from being copied or surpassed.

Banquet Speakers Honor Woolson

Breckinridge's Wit and Wilson's Presentation of the Air Fleet Please the Diners

THE aeronautic banquet held at the Park Central Hotel on Wednesday evening, May 7, was a joint event of the Metropolitan Section and the Aeronautical Chamber of Commerce of America. George A. Round, Chairman of the Section, stated that this was the last Section meeting of the year and announced the results of the election of Section officers for the coming year, as follows: Chairman, A. M. Wolf; Vice-Chairman, John F. Creamer; Vice-Chairman for Aeronautics, Charles Froesch; Treasurer, Erwin H. Hamilton; and Secretary, Grosvenor Hotchkiss. The retiring chairman expressed his appreciation of the work of Mr. Wolf as chairman of the papers committee and of that done by Mr. Hotchkiss as editor of the Metropolitan Section Booster.

Chairman Round then turned the gavel over to Henry S. Breckinridge, whom he introduced as a Kentucky lawyer who has been Assistant Secretary of War and a director of the Transcontinental Air Transport and is now known as the attorney and advisor of Charles A. Lindbergh. The toastmaster proceeded to call the roll of some of the notables who were present, asking them to rise for the inspection of the 200 diners.

T. A. T. Lawyer Introduces Notables

First came Eleanor Smith, among whose aviation records was averred to be a permanent one for pulchritude.

Next came Capt. "Eddie" Rickenbacker, hailed as the first ace of the American Aviation Service and Dan Shaeffer, to whom tribute was paid as standing for the far-sighted attitude of railroads in joining hands with aviation. Mr. Shaeffer is now chairman of the executive committee of the Transcontinental Air Transport. Major Leslie MacDill was introduced as chief of the Army Engineering Section at Wright Field. Chairman-elect Wolf came next and then Ralph H. Upson, who was introduced as the designer of the ZMC-2 airship. He was averred by the toastmaster to be visible in the "basement" of the ballroom seen in the mural decorations of the room. Under cross-examination, Mr. Upson testified that the aforementioned airship contained 3,000,000 rivets, and attorney for the prosecution pointed to Upson as the culprit who put them all in "and stands there none the worse for wear."

T. P. Wright was introduced as chief engineer of the Curtiss Aeroplane & Motor Co., then Major T. G. Lanphier, who is credited with maintaining a high degree of safety in aviation. The toastmaster said that he commanded the greatest training enterprise ever undertaken by the American Army at Issoudun, in France, a training field covering 52 square miles. When he assumed command, men were being killed at the rate of 17 per week. Only one death was recorded at that field during the last three weeks of the war, under Major Lanphier's command. Others who were introduced, with appropriate sallies, were Commander J. C. Hunsaker, E. T. Birdsall, Carl B. Fritsche and Giuseppe M. Bellanca.

Having thus exercised a few of the diners, Mr. Breckinridge paid his respects to George Washington and Herbert Hoover as the two engineer presidents of the United States, and then paid a feeling tribute to the memory of Lionel M. Woolson, placing him beside Wilber Wright in his contribution to the future of aviation. He enumerated the three obstacles to aviation as the weather, fire and pilot judgment. No panacea exists, he said, for the last, but public opinion and the growth of common sense will tend to restrain misguided action of too ardent pilots. Mr. Woolson's work on the fuel-injection engine has gone far to eliminate fires in the air, and his death was due to the weather, the other of the three great enemies of aviation. His death was characterized as a terrible price to pay for a dramatic demonstration of the validity of his life's work.

TOASTMASTER AND SPEAKER AT THE AIRCRAFT BANQUET

Henry S. Breckinbridge, Personal Counsel to Col. Charles A. Lindbergh, Toastmaster; and E. E. Wilson, of the Hamilton Standard Propeller Co., Who Spoke on Transportation versus Air Circus

Ex-Navy Expert Is Speaker

Com. E. E. Wilson was introduced as a man who has influenced the progress of the art of aviation much more than is popularly known, particularly by his contribution to the development of the now popular radial air-cooled engines and of the operation of Naval airplanes from aircraft carriers. Although Commander Wilson probably was the greatest expert in the Navy on the design of both airplanes and airplane engines, the toastmaster believes that Army and Navy officials are wrong who would restrict the resignation of such men to enter civil life. They become friends of the Service in industry, and make room for the promotion of other officers.

The aircraft squadrons of the battle fleet, which had flown that day over New York City, were taken as the text of the discourse of Commander Wilson, who is now connected with the Hamilton Standard Propeller Corp. The ten squadrons had been operating with the fleet in the Caribbean Sea all winter with their three mother-ships. Flying from Hampton Roads, where the ships were being overhauled, the airplanes were obliged to scatter to seven or eight landing-fields for accommodation, some of them going as far as Hartford, Conn. He represented these 150 airplanes as the largest single air force in the world today, and doubly valuable because of the mobility of their bases. They can go to the far corners of the earth, and could drop 100,000 lb. of explosives on any city.

Air Fleet Is the Spear-Head

Accepted only as an auxiliary to the Navy until recently, the air fleet is now recognized as the spear-head of the attack. Their effectiveness was strikingly demonstrated in the practice attack on the Panama Canal last January. In previous mock attacks, the fleet had bombarded the Canal under the protection of smoke screens, well knowing that no such attack could ever be really successful. On this occasion, the Saratoga started from a point 750 miles away from the Canal, 24 hours before the attack was due, escorted by a light cruiser, and delivered an attack by means of 85 airplanes in three groups. One group was a squadron of fighting planes capable of flying over any enemy squadron carrying 500 lb. of bombs, which attacked the two locks at Pedro Miguel and the Lake. This group had the good fortune to arrive at the moment when the defending planes had alighted to refuel. They drew the defenders away in pursuit after the attack. This attack was followed by heavy bombardment from 18 big torpedo planes approaching from three directions, each group escorted by pursuit planes and scouts. The third attack was made by two-seaters, which came in without opposition after a wide detour.

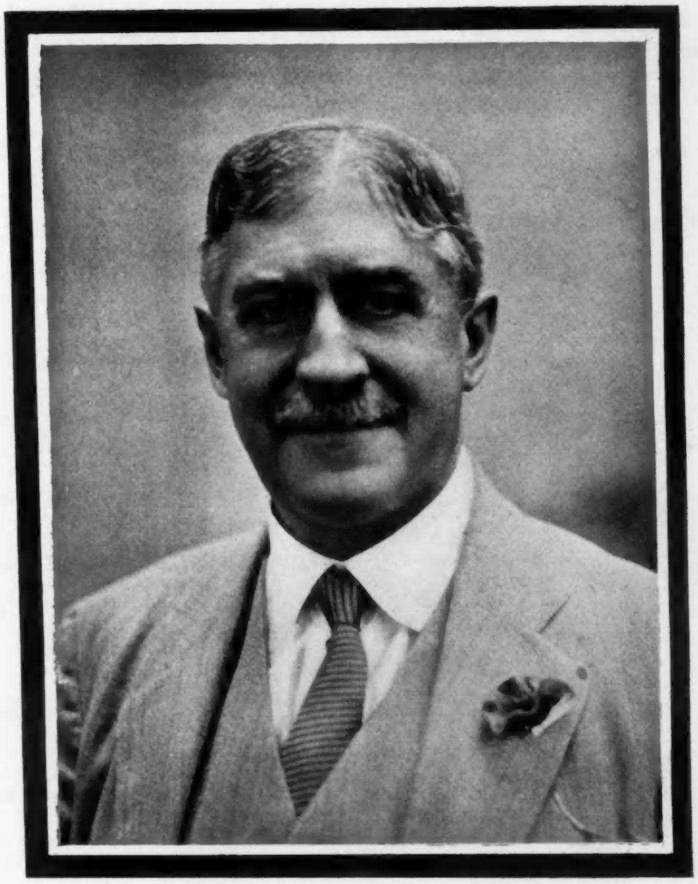
Commercial aviation began much later than military aviation, and has many lessons to learn, but Commander Wilson believes that they will be evolved readily. He believes that air transportation is sound; that it is undoubtedly the quickest, that it can be made the safest and most comfortable, and that ultimately it will be made the cheapest form of transportation.

In closing, Commander Wilson added his personal tribute to Mr. Woolson and suggested that the Packard Diesel engine on exhibition at the Aeronautic Show should be given a place in the National Museum along with other exhibits which symbolize the steps in the advance of aeronautics, in the memory of Lionel Woolson, who combined the outstanding characteristics of a real man and a real engineer.

Andrew Lawrence Riker

TO those members and guests of the Society who attended the Summer Meeting and saw and heard Mr. Riker speak at the Standards and 25th Anniversary Sessions, the news of his passing on the following Sunday at his home in Fairfield, Conn., was particularly surprising and shocking. He had seemed so well and vigorous at the meeting that no one would have doubted he would be privileged to enjoy many more years of beneficent life.

In the passing of Mr. Riker the Society not only loses its first President and one to whom, probably more than to anyone, it woes its existence, but all who have been associated or come into contact with Mr. Riker must feel that they have lost a real friend, for he was loyal to his associates and gracious to everyone. How Mr. Riker assisted in the organization of the Society and stood by it valiantly in its earliest and most precarious years is recounted in his own words delivered in an address at the Anniversary Session and in a letter over his facsimile signature in this issue of THE JOURNAL. These were his last services to the Society that he loved so well. And in the Historical Exhibition at the meeting he saw again the electric tricycle which he designed and built in the early '90's. When only 20 years of age he became president of the Riker Electric Motor Co., which introduced commercially the first toothed electric armature, and in 1895 he built a four-wheel electric motor-car. Four years later he established a straightaway one-mile record of 63 sec., which stood for 10 years, in an electric car of his own design.



ANDREW LAWRENCE RIKER

Thus Mr. Riker was not only one of America's earliest pioneers in automobile design and construction but, with other contemporary pioneers, including Charles Duryea, Alexander Winton and Henry Ford, he risked his life in speed trials and races to test and improve the engineering designs, materials and workmanship of motor-vehicles. Although his first vehicles were electrically driven and he so plainly manifested his interest in high-speed cars, his breadth of interest and his genius as an inventor and designer are indicated by the fact that Riker 5-ton electric trucks were among the earliest motor-vehicles operated in New York City. That was 30 years ago. He soon turned his attention to gasoline vehicles, however, and in 1902 designed the first gasoline vehicle built by the Locomobile Co. of America, which had been the largest builder of steam automobiles in the world. This first gasoline Locomobile had a sliding-gear transmission, a steel frame and high-tension ignition. Soon afterward he designed and built the Locomobile that won the Vanderbilt Cup Race on Long Island in 1903. For the pioneer work he did in the development of motor-cars the French Government awarded Mr. Riker a medal in 1900.

Mr. Riker was born in New York City in 1868 and received his education in mechanical and electrical engineering at Columbia University. He joined the Locomobile Co. in 1902 and later became vice-president and chief engineer of the company, a position which he retained until 1921, when he became vice-president of the Investing & Mfg. Co., of Bridgeport, Conn. From 1924 to 1928 he was engaged in the independent field as a consulting engineer, and in 1929 was elected vice-president of the Ventilouvre Co., in Bridgeport.

Mr. Riker was not only the first Member of the S.A.E., his application bearing the numeral 1, but he was President for the first three years, from 1905 to 1907. He was a member of the Finance Committee from 1910 to 1916, of the Ball and Roller-Bearings and Frame-Sections Divisions of the Standards Committee from 1910 to 1914, Chairman of the Miscellaneous Division in 1911, Chairman of the Electrical Equipment Division from 1912 to 1914, member of the Council in 1913 and 1914, Chairman of the Truck Division of the Standards Committee in 1917, member of the Constitution Committee in 1923 and 1924 and Chairman of that Committee in 1925.

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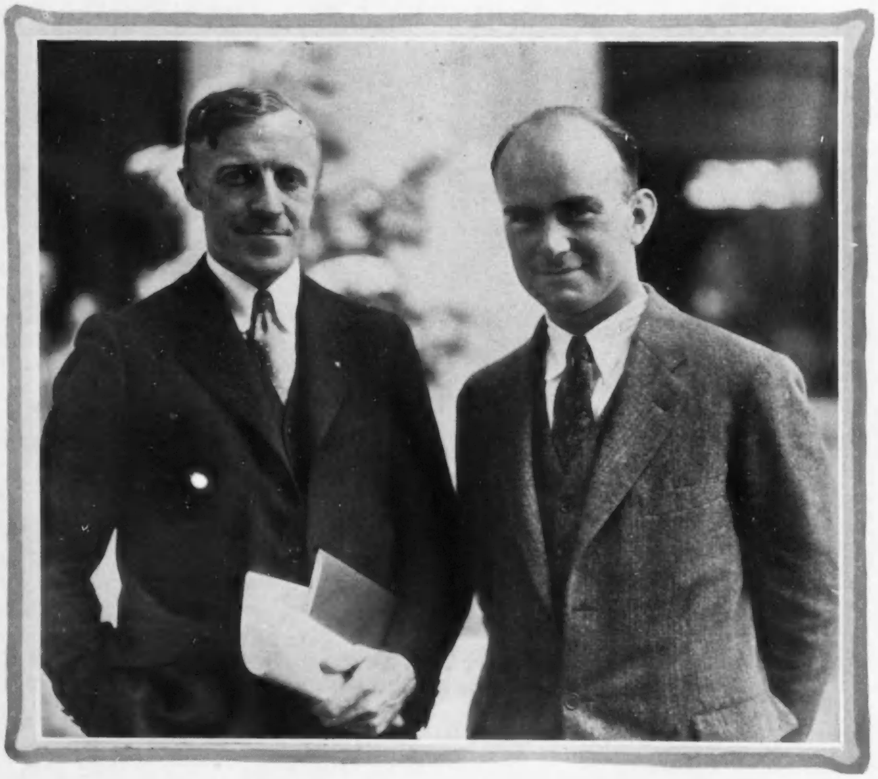
in some cases being as much as 20 deg. cent. (36 deg. fahr.)

Pumps No Sure Cure for Vapor Lock

The general conclusions of the tests were: that the use of a fuel pump eliminates most causes of vapor lock on the pressure side of the pump but introduces a serious tendency toward vapor lock on the suction side; careful designing of gravity systems, including the elimination of all unnecessary bends, use of sufficiently large tubing and avoiding changes in cross-section, can assist materially in reducing the tendency to vapor locking; and that weathering of the gasoline in the carbureter float-chamber tends to prevent vapor locking in the carbureter and may even enrich the mixture because of pressure developed above the gasoline. Dr. Bridgeman remarked that work had been done on only a limited number of fuel systems and that the investigation is to be extended to further variations.

After the Chairman had complimented the work of the Bureau on this problem, the first discusser was F. W. Heckert, of the V. G. Apple Laboratories, who had reverted to his reserve-officer status and flown from Dayton that morning with Lieut. E. A. Ross, who is an engineer of the powerplant branch of the Army at Wright Field. Lieutenant Ross also participated in the discussion. Ascertaining that the pump used in the experiments was of a type that is used almost exclusively on airplanes in this Country, Mr. Heckert reported experiments with another type of pump that has a much higher lift and has operated more than 3000 hr. with only a slight decrease in capacity. Dr. Bridgman said that such a pump might show better performance than the rotating pump.



AERONAUTIC-ENGINE DISCUSSERS FLEW FROM DAYTON

F. W. Heckert (Left) Was Connected with the Powerplant Laboratory at Wright Field for a Number of Years and Has Been Chairman of the Dayton Section During the Last Year. Lieut. E. A. Ross (Right) Is an Engineer in the Powerplant Branch of the Army Air Corps at Wright Field

Past-President Wall brought up the question of lagging the fuel line for heat insulation, and Harry F. Huf of the Atlantic Refining Co. reported having definite evidence that such insulation had been helpful; in fact, he considered it necessary on cars having thermostatically controlled radiator shutters that maintained a temperature of 180 deg. fahr. under the hood.

Many Airplanes Need Fuel Pumps

Lieutenant Ross told of flight experiments with a three-engine airplane having a gravity feed that normally gave about one-half of the 3-lb. per sq. in. pressure at the float chamber that was recommended by the carbureter makers. The side engines would stop alternately under some conditions and simultaneously under other conditions. Investigation showed that virtually no pressure existed at the carbureter when the engine failed at high altitude. The trouble has been remedied by applying fuel pumps.

Since the pumps were installed, Lieutenant Ross has flown the plane to Sacramento and had no difficulty with fuel flow at 15,000 to 16,000 ft., the highest altitudes attained. He considers the requirement for a pressure of 3 lb. per sq. in. to be greater than can be met by any airplane using the gravity system.

Installation Improvements Demanded

Arthur Nutt's paper was read, in his absence, by Arthur Leak, who is also attached to the Curtiss Aeroplane & Motor Co. Mr. Nutt's paper urged united cooperation instead of the present division of responsibility between the engine designer and the airplane designer in the installation of aircraft engines, and made suggestions for improvements of several sorts in such installations. The tubular rings upon which engines are commonly mounted are usually supported by structural members that are welded to the ring and attached to the fuselage at the four longitudinals. Inaccuracy is common in these structures, and many of them lack sufficient stiffness. Heavy forged or cast-aluminum machined rings sometimes are provided to protect the engine crankcase from the resulting loads. If a failure occurs, it will not be in an expensive crankcase, but the added weight is objectionable.

Four ordinary heat-treated alloy-steel bolts are frequently used for the engine mounting, sometimes fitting in reamed holes. The bearing area in the tubular structure is small, so looseness results soon. The remedy suggested is adequate bearing area and tapered-body bolts.

Precautions for Fuel and Oil Systems

Gravity gasoline feed is recommended for its simplicity, provided the

pressure head required by the carbureter can be secured, but Mr. Nutt reports having seen an installation in which the engine would operate as long as the airplane had its tail on the ground, but the engine would die as soon as the tail was raised during a take-off. The use of gasoline-resisting rubber hose with metal liners and the avoidance of sharp bends are recommended for the gasoline connections. A strainer of ample capacity with an accessible drain should be located near the carbureter.

The overflow from the pressure-relief valve is reported to be the cause of vapor lock sometimes, when it is merely a return to the pump inlet. A separate overflow pipe to the tank is recommended, but attaching the relief line at a point below the highest point in the discharge line guards against the accumulation of bubbles of vapor if it is necessary to make the return near the pump inlet.

The location of oil tanks often is such as to prevent effective cooling, and oil thermometers frequently are located at the engine oil inlet instead of at the outlet. With this arrangement, the maximum oil temperature may be 250 deg. fahr. when the instrument reads 200 deg. For new installations, thermometers are recommended by Mr. Nutt to indicate both the inlet and outlet temperatures of the oil.

Cowling and Mixture Heat Need Study

Too little attention is said to be given to carbureter heating. The atomized fuel leaving the carbureter should impinge against a surface heated to about 350 deg. fahr. Too much heat results in a loss in power, and too little heat causes poor carburetion and increased wear of the engine parts. The carbureter must be jacketed if warm air is not supplied sufficient to prevent the formation of ice around the jets and throttle. A properly designed carbureter heating system, according to Mr. Nutt, requires only a part of the exhaust from one cylinder.

Other insulation details that require more attention are the electric wiring, the engine controls and the cowling. Pure sheet aluminum, such as the production man favors, is not strong and stiff enough to withstand the pressure of the slip-stream, and duralumin may be no better if it is not properly annealed for forming and subsequently heat-treated. The development of an inexpensive, rigid and strong cowl fastener would be welcomed.

Cooperation Can Improve Installation

In discussing this paper, Mr. Heckert said that his eight years' experience with the Army Air Corps made him conscious of many shortcomings of the sort mentioned in the paper. There has been a constant battle between the Air Corps and the manufacturers to secure uniform and adequate installations. He recommended more direct cooperation between the Department of Commerce and the Army and Navy Services, because the latter have learned much that would be of great help to commercial operators. Mr. Heckert was afraid that many fliers would be frightened into early landings if they were aware of the outlet temperatures of the oil in their engines.

Further discussion had to do with carbureter heating and oil coolers, attention being called by Lieutenant Ross to the desirability of shutters to control the effect of oil coolers.

Vapor Lock Is Research Subject

Fuel Condition and Heat Around Carbureter Are Crucial —Acceleration Study Reported

PAST-President Bachman, in opening the Research Session on Thursday morning, spoke of the benefits in better mutual understanding that have resulted from the cooperation between the American Petroleum Institute, the National Automobile Chamber of Commerce and the Society of Automotive Engineers.

The first paper presented at the Session, that by C. S. Bruce, of the Bureau of Standards, was upon the subject of engine acceleration. It was a continuation of work that has been reported in previous papers by William S. James, Dr. H. C. Dickinson, S. W. Sparrow and other members of the Bureau of Standards. In presenting the paper, Mr. Bruce repeated some of the previously-given information and illustrations in regard to the laboratory apparatus and procedure.

Early tests were made in the laboratory, with only one or two engines. These were followed by tests of various cars on the road, made with the aid of portable spark-accelerometers.

The phase of the work particularly covered in Mr. Bruce's paper included comparisons of fuel performance with down-draft and up-draft induction systems and with three separate carbureters connected by short pipes to the three inlet ports of the engine. The latter arrangement is said to roughly represent cold carburetion, but complete parallelism with the other series of tests was not possible because the three carbureters had no accelerating-charge pumps.

How to Boost Acceleration

In concluding his paper, Mr. Bruce said that the effect of fuel volatility is shown to be qualitatively independent of engine design; that the effects on acceleration of increasing the heat applied to the manifold, the fuel volatility and the accelerating charge are similar, all three tending to increase the acceleration at low speeds; that heating the manifold or increasing the volatility of the fuel, if the mixture supplied is rich enough to give maximum power, will increase the acceleration at low speeds but decrease it at higher speeds. Properly proportioning the quantity and rate of injection of accelerating charge, together with a carbureter setting giving maximum power at constant speed, makes it possible to realize very fully the computed maximum acceleration over the entire speed range; and the total induction surface to be wetted is a determining factor in fixing the amount of acceleration charge required to produce optimum acceleration performance.

Answering a question by J. O. Eisinger, of the Standard Oil Co. of Indiana, Donald E. Brooks said that tests at the Bureau of Standards last year showed the effect of the accelerating pump. A number of similar cars of a popular make were used, one of them fitted with an accelerating pump. They were tried for acceleration, going through the gears. Cars without the pump required 6.2 sec. to travel 160 ft., while the one fitted with an accelerating pump could make the distance in 6 sec. flat.

Mr. Bruce remarked that no work has yet been done, to his knowledge, to determine the influence of the character of the interior surface of the manifolds, but that such a study has interesting possibilities.

Weathering and Vapor Lock

The effect of the weathering of gasolines on their vapor-locking tendencies was the subject of a paper by Dr. O. C. Bridgeman and Miss E. W. Aldrich, which was read by the latter, who is a junior chemist at the Bureau of Standards.

As an airplane reaches altitude, the temperature of the fuel in the tank remains higher than that of the surrounding atmosphere. Boiling frequently results from these conditions, indicating that a vapor-locking temperature at the existing pressure has been reached. If trouble from vapor lock does not occur before this boiling commences, according to the authors of the paper, it is probable that weathering will automatically prevent the occurrence of vapor lock after the boiling begins.

The work reported in the paper was undertaken to determine the extent of

weathering with a considerable number of gasolines under varying conditions, and the conclusion reached was that no appreciable change in vapor-locking tendency occurs during the short time that transpires before the vapor pressure of the gas-free gasoline exceeds the prevailing atmospheric pressure. After this condition is reached, the practical limit to the amount of loss by weathering and the change in the tendency to cause trouble by vapor lock can be quantitatively estimated for any given temperature and pressure. The conclusions are also applicable to motor gasolines and to certain phases of the problem of evaporation loss during storage.

Tests were made by heating the various fuels in a jacketed container with a connection to a manometer and a vacuum pump. The weight of the fuel was determined by weighing before and after the tests, and the loss was expressed as a percentage by volume. The quantity of fuel used in each test was 750 ml., in a container of larger capacity, simulating the condition in which an airplane takes off with tanks almost full, climbs to a given altitude and remains at that altitude for a given time. It is recognized that somewhat different losses might occur with other initial quantities of gasoline.

Vapor-Pressure Difference Significant

Among the conclusions and applications drawn are the following:

(1) The loss for a given time in a specific apparatus depends primarily upon the difference between the vapor pressure and the external pressure and only indirectly upon the actual temperature and external pressure.

(2) For a given pressure difference, the loss increases with the size of opening and with the time of superheating, up to a certain amount.

(3) When a gasoline is superheated at a given temperature and pressure, loss occurs until the vapor pressure of the residue at the temperature under consideration is reduced to the given external pressure.

(4) A series of gasolines maintained in a superheated state for sufficient time will ultimately have the same vapor-locking tendency, which depends upon the chosen temperature and pressure, but the losses from the various gasolines may be quite different. Vapor locking may be expected to occur before the gasoline has weathered sufficiently to change its vapor-locking tendency, particularly in the case of airplanes employing fuel pumps.

(5) If vapor locking does not occur before the gasoline tends to become superheated, weathering probably will automatically prevent its occurrence.

(6) The temperature at which the gasoline begins to superheat at a pressure of approximately one-half atmosphere, corresponding to 1800 ft. altitude, is approximately 25 deg. cent. (77 deg. fahr.) below the A.S.T.M. 10-per cent point. Therefore not much loss is to be expected from weathering in most cases with engines that are not supercharged unless the fuels are very volatile or are at an unusually high temperature.

Designing to Avoid Vapor Lock

Vapor locking was considered in the foregoing paper chiefly in a fundamental and theoretical way and as a function of the fuel. The final paper of the meeting, by W. C. Bauer, of the Standard Oil Development Co., considered the same subject more in its relation to actual fuel systems.

Installations of gasoline engines in any automotive service can be and are made so that vapor locking never occurs under any conditions of operation, according to Mr. Bauer. As some installations are troubled with gas locking under normal warm-weather conditions, the trouble seems to depend upon the design of the fuel system rather than upon the fuel.

Difficulties from fuel locking show themselves most commonly in failure of the engine to idle after a fast, hot run or in traffic; sometimes in intermittent or uneven acceleration after idling; sometimes during a silent high-speed run; and rarely cause complete stopping of the engine. Laboratory work has diagnosed the cause and location in the fuel system where each of these troubles has occurred, and the findings for six experimental fuels are compared in the paper with data obtained in road runs in hot weather.

Heat differential of more than 5 or 6 deg. fahr. between the vacuum tank and the carbureter is shown to result in vapor lock, regardless of the design. The design, however, has a great influence upon the horsepower and intensity of the vapor lock. When imperfect operation had been established during testing by heating the carbureter, heating the fuel tank to within 5 or 6 deg. of the temperature in the carbureter bowl would cause the trouble to disappear. Other arguments were offered for maintaining a relatively low temperature for the carbureter.

Preventing vapor locking by control of the fuel places a decided limit to the volatility of the fuel that can be marketed during the season of greatest sale. Better fuels are the easier to carburet and distribute uniformly and they give better acceleration and easier starting, according to the author. Current products of 17 different factories, to the number of 21 different cars and trucks and representing approximately

PARTICIPANTS IN THE RESEARCH SESSION

Top, Left to Right—Past-President B. B. Bachman, Chairman; Dr. O. C. Bridgeman and Miss E. W. Aldrich, Co-Authors of a Paper on Effect of Weathering in the Tank on the Vapor-Locking Tendency of Gasoline

Bottom—C. S. Bruce, Who Presented His Paper on Engine Acceleration, and W. C. Bauer, Who Gave a Paper on Vapor Lock

82 per cent of the Country's production, were tested at the Elizabeth laboratories for vapor lock. According to these tests, making the initial boiling point of fuel 10 deg. fahr. lower would not cause vapor locking on more than one-fifth of all the cars and would give a fuel that would be more desirable in many respects.

Cool Carbureters Could Use Better Fuel

Mr. Bauer's parting shot was that the enclosure of the engine seems to be ignored in the development of engines. He says that it is not unusual to be able to make a gain of 3 to 5 per cent in horsepower by simply raising the hood on the carbureter side of the engine. Both air and gasoline should be cool when delivered to the carbureter, heat necessary for vaporization being applied between the carbureter and the inlet valve.

Discussion of the last two papers, mostly the last, came after Mr. Bauer's paper was presented. A. L. Clayden, of the Sun Oil Co., reported that some testing he had done uncovered a case in which simply turning the carbureter around caused the temperature to drop 40 deg. in normal summer weather, suggesting that the position of the air intake is a most important factor. Mr. Bauer said that no changes in installations had been tried, in the Elizabeth tests, other than removing air heaters or other simple changes such as an owner might be likely to make.

Asked by Dr. Bridgeman how far the fuel-line temperature can be reduced, Mr. Bauer said there is no reason why fuels having a 10-per cent point of about 130 deg. fahr. cannot be made to operate perfectly.

Warm Winter Days Aggravate Problem

Dr. Bridgeman said that he knows of fuels marketed throughout the Country in the summer-time that have a 10 per cent point lower than 100 deg. fahr. He said that it is unfortunate in some ways that more casing-head gasoline cannot be used in the fuel, and that the elimination of vapor locking involves a compromise between the refiner and the car manufacturer. Mr. Beall admitted that much gasoline is sold that has too low a 10 per cent point, but gasoline must be made of a quality that will enable a car to start in the winter. During some winter days, the atmospheric temperature may be such that vapor locking may be serious.

Mr. J. P. Stewart, research engineer of the Vacuum Oil Co. interpreted Mr. Bauer's paper to mean that complete stoppage of the vehicle results from vapor lock in the line leading to the carbureter and that irregular running is caused by vapor lock in the carbureter itself. Mr. Stewart pointed to the filter as the most common source of vapor lock in the lines. This often is placed immediately above the exhaust pipe, and bubbles can be seen in the glass bowl of the strainer when vapor lock is encountered. He reported that one well-known car which has been troubled with vapor locking is now experimenting with an electrically operated fuel-pump placed in the gasoline tank.

Expedients for Design and Service

Service for cars that have been made in the past and are over-supplied with hot-spots must still be maintained, the discussers were reminded by R. E. Wilkin, of the Standard Oil Co. of Indiana. He suggested that vapor locking in the carbureter, during idling after fast driving, might be prevented by placing a thick asbestos plate between the carbureter and the heated riser, to reduce the flow of heat into the carbureter. Mr. Beall said that he knew that this suggestion had been effective on two or three models.

An auxiliary supply of high-test fuel was suggested by F. W. Heckert as a means for starting in cold weather and avoiding the necessity for a main supply that might be subject to vapor locking. Military aeronautic maneuvers in the Northwest last winter were aided by such a device, making use of a jar of ether and a hand-pump having a three-way valve. Dr. Bridgeman said that the operating range between the starting temperature and the temperature for vapor lock is approximately 140 deg. fahr. Cars equipped with thermostatic shutters operate at nearly as high a temperature in the winter as in the summer, and it is a perplexing problem to provide a gasoline that will make winter starting possible.

Foreign service was drawn upon by J. P. Kent, of Chrysler Motors, who said that trouble from vapor lock is solved in some hot countries by installing a metal shield to deflect away from the carbureter or fuel pump the air heated by the exhaust pipe. A suggestion made by B. D. Mears was that the intake air might be drawn from the driver's compartment instead of from under the hood. Chairman Bachman questioned the result in case of a backfire.

Truck and Coach Problems

State Restrictions on Width, Together with Low-Pressure Tires, Puzzle the Designer

TUESDAY night's session on motortrucks and motorcoaches was honored in having as its chairman one of the earliest Presidents of the Society, H. W. Alden, chairman of the board of the Timken-Detroit Axle Co., who was eminently qualified in all respects for the job. In his introductory remarks leading up to the first paper to be presented, he indicated some of the problems that now confront the designers of heavy-duty motor-vehicles and the parts for them.

Chairman Alden stated in effect that the difficulties of the motor-truck and motorcoach builders are very considerable, for, whereas we used to have vehicles that weighed 18,000 to 20,000 lb. and traveled at speeds of about 30 or 35 m.p.h., which was thought to be a rather stiff performance, we now have motorcoaches designed for 39 or 40 passengers into which it is not uncommon to crowd 110 persons and which are equipped with 150 or 160-hp. engines to drive them at 30 or 40 m.p.h. in cities. Then we have to be able to stop these vehicles, with their enormous momentum, which is quite a problem. These vehicles weigh 29,000 to 30,000 lb., and with 105 to 110 passengers aboard, the gross weight is up to 31,000 or 32,000 lb. Twenty-inch-base tires used to be entirely adequate to give good braking performance in cities, but to stop the present motorcoaches with a set of four brakes inside of 20-in.-base tires has become increasingly difficult. Safety is the most important factor to consider, and to provide adequate brakes inside of tires of that inside diameter presents a serious problem.

How Tires Affect Axle Design

The difficulties confronting the designer of heavy-duty vehicles because of the over-all width restrictions imposed by the various States on the one hand, and by the very large section low-pressure tires on the other hand, were pointed out in detail by L. R. Buckendale, of the Timken-Detroit Axle Co., who read the first paper, after which the series of lantern slides accompanying it were shown and general discussion of the subject followed.

Some of the factors influencing truck design were the adoption of the 56-in. standard vehicle tread, the over-all width restrictions, the adoption of high-pressure and later of low-pressure pneumatic tires, improved roads, increased engine-power, increased vehicle capacity and weight, and increased speed. To cope with some of these conditions, the designers brought out dual rear tires, smaller diameter and wider brake-drums, front-wheel brakes, inclined the steering-knuckle pivot, re-

duced the overhang of the steering arm cross-rod beyond the ball joint, and developed the six-wheel vehicle with tandem single rear tires. Now, with the demand for still greater carrying capacity, dual low-pressure tires are being placed on the four rear wheels of the six-wheel vehicles and on the front steering wheels as well.

Parts Cramped by Tires

The speaker then proceeded to show just what the designing of a truck to come within the over-all width of 96 in. and within the maximum gross load of 24,000 lb. imposed in some States means as regards the rear axle and the brakes. With six low-pressure tires, each carrying 4000 lb., a 10.50-20-in. size is called for and the duals should be set, according to the tire engineers, 12¾ in. apart from center to center, making the distance between the inner tires 48 in. Within this distance must be found room for the driving mechanism, the vehicle springs and the brakes, with clearance for the parts. The springs and their clips reduce the available width for the chassis frame to 34 in.

To stop a 24,000-lb. load operating at passenger-car speed requires an effective brake-drum diameter of 17¼ in. and a width of 5 or 5½ in. Such a brake diameter inside a 20-in.-diameter tire leaves a clearance of only 1⅜ in., in which must be found room for the brake-drum thickness, the rim for the tire, means for holding the rim to the wheel, the tire valve and space for the flow of air to carry off the heat generated by the brakes. The real problem, then, is to design brake-drums of sufficient stiffness to resist distortion and capable of absorbing and dissipating the large amount of heat generated by the high pressures set up by the shoes of self-energizing or power brakes. The only answer seems to be, according to Mr. Buckendale, to increase the inside diameter of the tire used on vehicles of more than a certain weight, and he suggested a 22-in.-base size for tires of 4000-lb. capacity or more.

Solution Requires Cooperation

Front-axle problems have also been introduced by the low-pressure tire, which requires more space in the extreme-turn position so as to clear the frame and steering connections. And the 34-in. frame, with its flanges, creates the problem of finding room for the larger engines required to move the vehicle at the speeds made possible by the low-pressure tire.

The foregoing considerations bring out the fact, according to the speaker, that much greater cooperation is going to be necessary between all the engineers responsible for the development of tires, brakes, axles and the vehicle as a whole if the development is to go forward toward the ideal vehicle without burdening the industry with excessive costs. The time has come, he asserted, to do away with the arbitrary laws fixing the characteristics of motor-vehicles and leave the determination of what may be operated over given roads in the hands of engineers employed by the States. These State engineers should be guided by a general code that would meet the varying geographical conditions and the road limitations. Such a code could be prepared by automotive engineers working with the Bureau of Standards and operating through the departments and commissions provided by the United States for this purpose.

Tire Makers Defended

Following the showing and explanation by Mr. Buckendale of the slides accompanying his paper, the discussion dealt mainly with the tire makers' responsibility for the difficulties and the changing or abolishing of the 96-in. width and other specific legal limitations. B. J. Lemon, of the United States Tire Co., said there is no doubt that the laws should allow a maximum width of 106 or 108 in., and that such a figure is going to be asked for, particularly on super-highways between large cities. A movement that has already been started to influence State legislators to recognize the requirements is going to have its effect in the next year and a half so that vehicles can be built to take dual tires of the 12-in. size at least.

Although the demand for change-over to low-pressure tires seems to be the result of propaganda by the tire makers, the demand is coming more from the users than from the tire makers, according to Mr. Lemon, although the latter know the advantages of the balloon tire and competition is so keen that they are trying to educate the vehicle owner to get a correct change-over.

In this connection, K. D. Smith, of the Goodrich Rubber Co., asked Mr. Buckendale if, in changing from solid or high-pressure tires to low-pressure tires, any ill effects are produced on axles originally designed for the smaller-section tires. Mr. Buckendale answered that such a change-over increases the overhang on the bearings and the static bending load on the axle structure. This is compensated for to some extent by the lower impact values of the low-pressure tires, and the torque mechanism derives some cushioning effect and torque reaction of the tires.

Opposed to Recommending Dimensions

For the very comprehensive presentation of the subject in his paper, Mr. Buckendale was complimented by F. C. Horner, of the General Motors Corp., who said he was particularly interested in the last paragraph relating to developing a code of restrictions by the automotive engineers in cooperation with Federal Government bureaus and departments. He repeated a warning that he has sounded several times before at meetings of the Society against the danger of recommending any specific size and weight limitations, as no one can predict what the future requirements of commercial vehicles will be. For that reason he is opposed to the proposed motorcoach code, which contains limitations that are already unsound.

Regarding brake-drum size and the suggestion to go to 22-in.-base tires, Duncan P. Forbes, of the Gunite Corp., stated that a larger brake-drum is needed to stop a vehicle mounted on 22-in.-base tires than on 20-in. tires, hence very little is to be gained by adopting the larger size. Instead, he feels that the best results can be obtained by reducing the brake-drum diameter to 16 or 16½ in. and using suitable brake-drum and lining material to give satisfactory results, thereby providing room for air circulation.

Referring to Mr. Horner's remarks, A. J. Scaife, of the White Motor Co., said that anything that is done toward removing or changing the legal limitations should be done in complete cooperation with the lawmaking bodies, which undoubtedly hold the whip hand. The best method, he believes, is to reach agreement through such meetings as the present one regarding the over-all width required and then use propaganda to bring about the necessary legislation. The Society itself should keep aloof from legislation, because it is very undesirable to have any limiting dimensions incorporated into the laws, but it is desirable to have regulations that can be changed from time to time by the State highway engineers. The National Automobile Chamber of Commerce is interested in the matter and the Society probably will have to work through it. Some united action should undoubtedly be taken so that the several States will take action toward similar regulation.

The very narrow highways that exist extensively now have an important bearing on the question of increasing the over-all width limitation, said Mr. Scaife, and, although United States four and six-lane highways undoubtedly will come in time, the present time is probably a little too early to get the desired action. The immediate question, as he sees it, "is what can be done to conform with the 96-in. limitation; possibly we shall have to put the spring seat on the differential and use a monorail for a frame, with a Sperry gyroscope to balance the load."

Thinks Tire Equipment Stabilized

A ray of hope was injected into the discussion by J. E. Hale, of the Firestone Tire & Rubber Co., when he stated his judgment to be that the

tire situation is going to be stable for a long time and that it is not likely anything new in tire equipment for trucks and motorcoaches will come for a considerable period. He expressed his belief that a very satisfactory range of tire sizes to cover the needs of the industry have been chosen, and that the parts and vehicle designers can go ahead with the feeling that the tire makers will not interject any new problems.

Some way to dissipate brake heat must be found to keep down the temperature of the tire beads. This is being worked on, said Mr. Hale, and an investigation is under way to determine the most effective way of dissipating that heat. At the same time the tire makers are trying to arrive at a figure expressed in degrees fahrenheit above which the tire beads should not be heated.

Practical Limitations also Enter

Mr. Buckendale was complimented on his paper also by B. B. Bachman, of the Autocar Co., who thought it desirable to add emphasis to what the author had pointed out regarding the importance of the effect of tire development upon all of the details. He does not blame the tire companies for the direction in which their development has led them, but some individuals in large commercial organizations who may not be fully posted on technical matters have been over-optimistic and over-zealous in advocating applications, thereby causing irritating embarrassment.

There are some practical as well as legal limitations to the width and length of vehicles. Mr. Bachman agreed with Mr. Horner that the placing of definite limitations of dimensions, particularly in the present period of development, will become embarrassing because it is almost impossible to predict where the trend of development of high-speed heavy-duty vehicles will ultimately lead. However, we must recognize the practical limitations imposed by the fact that the heavy-duty vehicles are not operating alone on the highways, that they must get into restricted places to receive and discharge cargo, and that tires of a given size require certain weight limitation.

Reverting to the statement made by Mr. Smith that, as the tire-base diameter is increased, the duty on the brake is also increased, Mr. Bachman referred to Mr. Buckendale's statement that as the brake size is decreased the superficial area of the brake to dissipate heat and the volume of metal to absorb heat are also decreased. The industry can decide which is the lesser of the evils. The engineers are struggling to find the answers to this question and to the problem of stability on narrow frames, the steering geometry and associated problems. The solution of these calls for the cooperation of the tire, axle, brake and other parts makers, the vehicle builder and also the highway engineers.

Balloon Tires a Godsend

Chairman Alden remarked that he believed the vehicle and parts makers would substantiate the statement that they bear no grudge against the tire men, who have brought along a pneumatic tire that has been a godsend to the industry; if we had not had the pneumatic tire we would have had no truck business. The present combination of elements does not work very well together, however, and he is sure that the tire maker is as anxious to solve the problems as are the vehicle and parts makers. With the cooperation of all concerned, the speaker said he is sure that the industry will pull itself out all right in the end.

PRINCIPALS AT THE MOTOR-TRUCK AND MOTORCOACH SESSION

Left to Right—Past-President H. W. Alden, Chairman; A. J. Scaife, Who Presented His Paper on the Speed and Ability Factors of Coaches and Trucks; and L. R. Buckendale, Who Gave His Paper on the Effect of Low-Pressure Tires on Axle Design

Speed and Ability Factors

Prior to giving his paper on Analyzing Speed and Ability Factors, Mr. Scaife took occasion to remark, as Vice-Chairman of the Motor-Truck and Motorcoach Division, that this was the first truck and coach meeting the Society has had since the S.A.E. was reorganized on the lines of the various activities, and he was gratified by the attendance at the session, considering the number of members directly interested in heavy-duty vehicles compared with the number concerned with passenger-cars.

Although the subject of his paper is a broad one, he dealt with only one phase of it that interested some of the engineers, especially the operators. This is the ability of heavy-duty vehicles to climb the steepest grades encountered on the highway. He referred to Dean Kimball's address at the Annual Meeting last January on the economics of production, in which the law of diminishing returns was pointed out. This same law is applicable, in Mr. Scaife's opinion, to the motor-truck and motorcoach as regards the most desirable size for maximum financial return on the investment, and he believes that we are approaching the point where diminishing returns are beginning to be realized. He instanced the double-deck bus, which has not been the revenue producer it was expected to be in city service, as indicating that a point exists beyond which it is not profitable to increase the carrying capacity of a single vehicle.

Truck operators are today asking for trucks, equipped with pneumatic tires, which will travel at high speed carrying loads in excess of 37½ tons, gross, over grades of 10 to 20 per cent. The problem is to find what tne maximum gross weight should be to get the greatest revenue from the investment. When the loads to be carried are greater than the physical and legal limitations permit on a single vehicle, the next step is to use four-wheel trailers or a tractor and semi-trailer combination and four-wheel trailers. However, in any case the speed and the ability of the power vehicle must be given serious consideration. It is believed to be desirable for the operator to have a survey made of the routes over which the vehicles are to operate, taking into account the maximum grades and whether the loads are to be passengers or merchandise, so that a type of vehicle will be selected that will have sufficient ability as to both power and speed.

Study of Climbing Ability

To illustrate, a series of charts showing what can be expected of vehicles of a specified rating provided with predetermined tire sizes and gear ratios were shown and explained. These showed the maximum grade the vehicle can negotiate in the different transmission ratios and also the ability and speed in each ratio. They revealed at a glance what the operating conditions of the vehicle would be over a certain route and whether the vehicle would give the performance desired by the operator. They showed also whether the engine had sufficient ability, whether the gear ratios in transmission and axle are satisfactory for speed and power, and whether the load carried is excessive.

Digressing for a moment from his paper, Mr. Scaife said that many good men in service stations spend a great deal of time, energy and money in trying to recondition a vehicle to do work that it was not designed to perform, simply because the problem has not been analyzed. They think that by grinding the valves or setting the spark-plug or changing the ignition they can get more power out of the engine when the power is not there. They do not want to change the gear ratio because that interferes with the direct-gear performance. It is surprising, he said, what some men who ought to know better, will try to put over.

Going to the blackboard, the speaker drew a characteristic horsepower curve, which he said will show probably the maximum around 2200 to 2400 r.p.m., and a torque curve with the maximum around 1000 r.p.m. Although we all talk horsepower, the engine cannot be kept up to 2200 r.p.m. on a grade without a big powerplant. A vehicle has just so much ability, and this can be used in three ways: (*a*) for speed, (*b*) for heavy work, or (*c*) partly for speed and partly for heavy work.

Chairman Alden, just before adjoining the session, said that "the problem presented by Mr. Scaife is an extremely important one and that the curves he drew are the nub of the whole thing; if you keep that matter of torque in mind all the time and forget about the horsepower, you will get a great deal farther in solving your traction problems and also your structural problems all the way through."

Methods of Maintenance Compared

Self-Service versus Service-Station Maintenance Debated at the Transportation Session

F. C. HORNER, assistant to the vice-president of the General Motors Corp., was chairman of the Transportation Session held Wednesday morning, May 28. H. V. Middleworth presented the only paper, entitled Self Maintenance as Compared with Service-Station Maintenance; but this was supplemented by voluminous prepared discussion by authorities on the broad subject of transportation, as well as by oral discussion from the floor.

Maintenance Methods Compared

Mr. Middleworth remarked that the conditions under which motor-vehicles operate are so varied that it is extremely difficult to arrive at any fixed or definite rule governing the method of maintaining motor-vehicle fleets. Further, that it is difficult to determine what the minimum number of vehicles should be to justify the establishment of a self-maintenance organization. We speak of large-scale operators and small-scale operators, but just where to draw the line between the two is without doubt a difficult problem, he said.

Suppose we say that a large-scale operator is one who operates 40 vehicles or more and a small-scale operator is one who operates less than 40 vehicles, said Mr. Middleworth. It is obvious that the large-scale operator with equipment separated into small fleets in remote localities would bring his problem into the same class as that of a small-scale operator. With such a condition existing, it seems reasonable to assume that "service-station maintenance" would be more economical considering miles traveled, time out of service and the like, which really should be considered. The successful and economical operation of a fleet of motor-vehicles depends very largely upon the extent of its actual use in the capacity for which it is intended. Let a motor-vehicle cease to function for any cause whatever, and at once it becomes an item of expense. Therefore, any legitimate means which can be employed to eliminate time out of service is a further step toward its successful economical operation.

Before any definite conclusion is reached regarding the advisability of service-station maintenance, Mr. Middleworth continued, a careful study should be made of the conditions under which the fleet operates. If it is found that it is broken up into small fleets of say 40 vehicles or less, operating in several separate and remote districts and necessitating many miles of travel to and from a central repair-station, then, from an economical viewpoint, self maintenance would seem out of the question. If after careful study it is decided that service-station maintenance is desirable, a thorough survey should be made of the service-station facilities available in the districts from which the vehicles operate. In this connection it is generally understood that service-station managers are glad and usually eager to get work from responsible concerns. They also are willing to allow a satisfactory discount on work done for such concerns. It is understood that work of this nature frequently is given precedence over other work that is considered of less importance. Arrangements can be made with service stations to do work at night, thereby eliminating the necessity of laying up the vehicles for repairs during the day. Arrangements also can

be made to have the painting done over week-ends so that time out of service can be reduced to the minimum.

Many large operators maintain central repair stations equipped with modern machinery and appliances necessary to perform the work required in maintaining their fleets. Speaking of his own company, Mr. Middleworth said that during the past five years it has maintained a central reconditioning station successfully and economically. He then described briefly its organization, equipment and operation.

Selection of Personnel

In summing up the situation, Mr. Middleworth remarked that the vital factor affecting the successful and economical operation of an automobile repair-station is the careful selection of the personnel. If an operator has a sufficient number of vehicles so centralized as to justify the establishment of a self-maintained repair-shop and if he employs a supervisory force equal to that of the manufacturer's service-station, there can be very little doubt that self maintenance is more desirable than service-station maintenance. If, on the other hand, an operator has not a sufficient number of vehicles so centralized or, if a greater portion of his fleet is operating from several separate and remote districts which would necessitate many miles of travel to and from a central repair-station and an unreasonable period of time out of service, it seems reasonable to consider service-station maintenance the more desirable.

Prepared and Oral Discussion

Chairman Horner remarked that self maintenance as compared with service-station maintenance has been a real problem and still is and always will be such to the operator of motor-vehicles. We have been dealing with this subject in the Operation and Maintenance Activity for some time, he continued. Mr. Middleworth has made a very close study of this problem and has dealt with it in the Transportation Activities Committee. We are now to hear the prepared discussion and, afterward, I hope there will be a full discussion from the floor.

Centralization or Decentralization?

Because operating conditions are so varied, Martin Schreiber, of the Public Service Coordinated Transport of New Jersey, said that he agrees with Mr. Middleworth that the maintenance plan adopted should be determined by a careful study of each operation rather than by any fixed rule. An important point has been raised regarding the self maintenance of motor-vehicle equipment, he continued; that is, is centralized maintenance more desirable than decentralized maintenance? He remarked also that the London General Omnibus Co., which operates about 5500 motorcoaches, probably the largest transportation fleet in the world, is an outstanding example of centralized maintenance. On the other hand, Mr. Schreiber's company, which operates the largest fleet of motorcoaches in America, according to his statement, follows the decentralized-maintenance plan.

The principal advantages of decentralized maintenance, Mr. Schreiber stated, are the reduction in the time that vehicles are out of service, elimination of non-revenue mileage to the central repair-shop, and the placing of responsibility where it belongs; that is, the local garage-man, who is directly responsible, is allowed to decide how and when the repairs are to be made. With the decentralized plan in effect, a definite inspection system of routine adjustments and repairs must be made. This maintenance system should be somewhat flexible, however, as it is found that the most economical maintenance is secured by varying the repair operations to suit the different operating conditions existing.

In conclusion, Mr. Schreiber stated the advantages of decentralized maintenance to be:

(1) Elimination of divided responsibility for maintenance

(2) The maintenance responsibility rests with the person who services and operates the vehicle

(3) The local garage-man's personal knowledge of each vehicle allows him to effect economies in maintenance not otherwise possible

(4) The time a vehicle is out of service for repairs is reduced

(5) The spirit of competition existing between the various garages results in strenuous efforts to reduce maintenance costs and eliminate road failures

(6) Non-revenue mileage to a central shop is largely eliminated

In the absence of Mr. Schreiber, his prepared discussion was read by A. A. Lyman, of the same company.

Outline of Satisfactory Maintenance Plan

The discussion by F. B. Whittemore, manager of service promotion for Mack Trucks, Inc., was read by M. C. Horine, of the same company.

Mr. Whittemore stated that, conceding that individual owners and small-scale fleet-operators secure the best, most economical and most satisfactory maintenance from service stations, he affirms that operators of more than 40 units in one district should do their own ordinary running repairs and adjustments and, in some instances, unit-assembly exchanges. For more pretentious repairs, the choice between central repair-shop and service-station maintenance rests on the convenience and dependability of available service-stations.

For general repairs and unit-assembly rebuilding, said Mr. Whittemore, operators having less than 200 vehicles in one district will serve their best interests by intrusting this work to a factory-branch service-station employing specialized mechanics under superior supervision, having a complete stock of replacement parts, equipment and time-saving precision-tools; that is, a unit of a service organization operating on the principle that quality parts and quality workmanship at reasonable cost are essential elements of lasting and satisfactory service.

Continuing, Mr. Whittemore stated further that operators must arrange for the garaging of their fleet in a public garage or in a building either rented or owned. Ordinary running repairs, adjustments, inspections and periodic lubrication call for little or no additional space beyond that required for garage purposes and, consequently, little additional expense. The availability of service-station facilities should be investigated carefully before the fleet operator decides how major repairs are to be accomplished. The operator should then count the cost of establishing and maintaining a central repair-shop requiring additional floor space, machines, tools, equipment, a stock of replacement parts, and a crew augmented by skilled mechanics and supervisors. Such surveys may indicate in some instances that the operator's best interests will be served by relying upon service stations for all general repairs, and, in others, by maintaining a central repair-shop.

In conclusion, Mr. Whittemore stated that factory branch service-stations can provide quality workmanship, can do work quickly, can do dependable work at minimum charges, can operate with less overhead in regard to labor, and can take care of peak loads by calling upon other stations within their organization.

Fleet Organization and Methods

After saying that, so far as the major problem of self maintenance versus service-station maintenance is concerned, from the viewpoint of motor-truck maintenance, F. K. Glynn, of the American Telegraph and Telephone Co., said that there is no difference or dividing line because service-station maintenance should be the most economical for a one-vehicle or for a 1000-vehicle fleet. He then asked the following questions:

(1) Speaking to the fleet operators, are the companies for which we work in the automobile business or are they selling dry-goods, coal, gasoline, electric power, public-utility service, and the like? That is, are our companies specialists along motor-vehicle lines?

(2) Has any one of us as many vehicles in his fleet, of one particular manufacture in one locality, as would be serviced regularly by a progressive dealer for that manufacturer?

(3) Can we afford a relatively large per-vehicle investment in shop building, ma-

chinery, special tools, spare parts, spare units and the like?

(4) Does a fleet operator with an extensive shop miss seeing the woods because of the trees for the reason that he is too interested in making his shop pay and is neglecting the economies of the road-running of his vehicles?

Mr. Glynn then showed lantern-slide views giving tabular data; one was a summation of various different types of 500-vehicle fleets wherein the data were comparable, and the other being a typical organization for a 500-vehicle fleet that uses manufacturers' and commercial repair shops. As to the duties of inspector repairmen, he referred to his previous paper, which was published in the S.A.E. JOURNAL, August, 1929, p. 148.

Continuing, Mr. Glynn made comments at length upon the history of service stations, how the keeping of the vehicles on the road satisfactorily can be made to tie-in with a commercial repair-shop, and the like. In conclusion, he summarized what seems to be the present conception of the job of the motor-vehicle supervisor or fleet operator, as follows:

(1) Build up an efficient operating organization

(2) Determine the most efficient type, make and size of chassis, body, accessories and auxiliary equipment for each job

(3) Determine the economic time for replacement

(4) Arrange for the purchase of all automotive equipment, maintain an adequate stock of parts and supplies, and arrange for the disposal of replaced equipment

(5) Determine the practices to be followed in the daily maintenance and in the repair of vehicles

(6) Design layout-equipment and locate garages to house the fleet

(7) Collect, compile and study operating data to effect improvements and economies

(8) Measure operating results

(9) Study the use of the vehicles in the field with a view toward increased productivity and the elimination of lost time

(10) Determine the qualifications of the drivers and train them in the safe operation and efficient care of the vehicles

(11) Review proposed motor-vehicle laws

(12) Handle registrations and licenses

Shall the Two Systems Be Combined?

O. M. Brede, director of service for the General Motors Truck Co., said in part that he believes the correct answer to the problem lies in the combination of the two systems. We must necessarily keep in mind, he continued, that the problem involves small, medium and large fleets, centralized and decentralized fleets, fleets standardized on one or two makes of vehicles and fleets in which there are many makes of vehicles. He remarked that there are operators at present equipped for and operating practically 100 per cent self-maintenance, others partially equipped and operating partial self-maintenance, and a third group that is not equipped and is using service-station maintenance.

In Mr. Brede's opinion, any attempt to set up certain rules by which to definitely classify operators and in such a manner determine for or against either self-maintenance or service-station maintenance would, he feels sure, fail. To determine the most economical system of maintenance, an individual analysis should be made of each fleet, taking all factors into consideration.

Mr. Brede went into considerable detail regarding small, medium, and large-size fleets, saying that the small fleets include large-scale operators having widely decentralized fleets, such as an operator who has, say, 400 vehicles operating in 50 different cities. He feels that the three systems referred to in the foregoing each has its suitable method as regards maintenance.

Low Maintenance-Costs the Incentive

H. C. Marble, of the White Motor Co., said that until some unborn magician pulls from his hat a motor-vehicle which will function perfectly without attention for its allotted period of years and then disintegrate as a unit so that the debris can be shovelled off the road and onto a junk pile, thereby solving all problems of parts, repairs and trade-in of used cars, we are forced to consider maintenance problems as they actually exist. The fleet operator and the manufacturer have an identical interest through a common desire for low maintenance costs; in efficient and economical maintenance they are closely linked as are Siamese twins. The small-scale or the large-scale operator with a widely scattered fleet had best employ the facilities of the service station than to attempt self-maintenance. The operator of a large concentrated fleet can profitably install a system calling for some degree of self-maintenance, but the question as to whether or not this system should embrace every variety of repair is open to argument. As to overhauls, there may be variance of opinion as to the most profitable procedure.

Operators using replacement of major units should remember that the cost is based upon the cost of bringing the unit removed up to standard, rather than on a fixed cost per unit. If unit replacements do cost slightly more, it usually will also be found that this extra cost is more than made up by the lessened time the transportation unit is out of service. Service stations are fully equipped to handle major unit-repairs more efficiently because their volume of such business justifies their having the necessary personnel and equipment. The question of whether the fleet operator can or cannot save money by self-maintenance is largely a question of volume of major repairs.

The operator who is endeavoring to decide between self-maintenance and service-station maintenance should first determine that self-maintenance will or will not bring to him a saving in actual dollars. If it will, he should ascertain whether the amount of the saving is a return upon the capital investment required commensurate with the return which would accrue to him through the employment of a like amount of capital more directly in the line of his own business.

Comparisons Between the Two Systems

A. H. Gossard, of the Middle West Utilities Co., Chicago, said that the subsidiaries of this company operate in 30 of the 48 States, all being east of the Rocky Mountain States. Each subsidiary company is regarded as a self-contained operating unit under a decentralized plan of operation, with the executive head held responsible for results. Operation of automotive equipment comes under whatever plan appears to be the most feasible for obtaining the best results. Many subsidiaries have no company garages, while others service all automotive equipment with company garages.

To determine whether there is any advantage in maintenance by either one of these methods, or the other, he showed tabular data to compare the operating costs of automotive equipment in a number of the subsidiary companies that are operating garages with subsidiary companies that have no garages.

These figures do not show that companies using service-station maintenance entirely obtain any better results than companies having one or more company garages, Mr. Gossard continued. The company showing the lowest operating cost maintains the equipment by a company garage. The equipment, though, is limited to a few makes and is operated on a high load-factor and over a long period. Close supervision makes this possible. Generally, he stated, it is our accounting practice to conform to the practice of automotive dealers in servicing equipment. Charges are made for repairs to equipment on a basis that includes the overhead of conducting company garages in the labor charge; for example, the labor charge, if costing 75 cents per hr., is charged for at the rate of $1.50 per hr. It is not our practice, though, to have company garages show a profit. If the doubling of the labor charge at the end of the year shows a credit to the garage, an adjustment is made in the labor charges. The difference between the cost of labor and the charge for labor is carried in a suspense account, which is balanced with all indirect charges to the garage.

On account of control of service, self-maintenance is the most satisfactory, in Mr. Gossard's opinion. Either self-maintenance or service-station maintenance calls for supervision. Too much or too little service can come from either method. It is much easier to service and control self-maintenance and, for this reason, we believe it will

prove the most satisfactory, all things being equal.

Criticism on Lack of Progress

Capt. Walter C. Thee, U.S.A., stated that, after hearing various papers read on maintenance of motor transportation by various members of the Society who are in charge of the maintenance of large motor-vehicle fleets or factory service-stations, in his opinion thé systems that are employed are practically identical, and the fundamental principles are similar to the average service-station of 10 to 15 years ago. With the exception of modern shop-equipment installed in the present-day service-stations, very little improvement has been

Principal Participants in the Transportation Session

(1) H. V. Middleworth, Who Presented a Paper on Fleet Maintenance. (2) M. C. Horine, Who Read Prepared Discussion by F. B. Whittemore and Took Part in the Oral Discussion. (3) F. C. Horner, Chairman of the Session and of the Transportation and Maintenance Committee. (4) T. C. Smith, (5) H. C. Marble and (6) O. M. Brede, Discussers

made on the system of maintenance; that is, very little has been done to apply to maintenance of motor-vehicles fundamental principles or laws of management that have been developed in industry during the last 10 to 15 years. He believes that some of the S.A.E. members will agree with him that maintenance of motor-vehicles has been stagnating and has progressed not nearly so fast in regard to economy and efficiency as has the production of motor-vehicles.

The cause of this stagnation, said Captain Thee, can be explained by saying that mass production has not been applied to the repair of unserviceable unit-assemblies, and economic factors applying to mass production have not been considered in the repair of motor-vehicles. He cited the paper by John Younger, published in the S.A.E. JOURNAL, December, 1928, p. 568, in which it was shown how efficiency and low production-costs have been attained in the production of motor-vehicles by following certain fundamental principles of manufacturing management based upon economic laws. He also referred to his paper presented at the S.A.E. Transportation Meeting in Chicago, Oct. 25, 1927, and published in the S.A.E. JOURNAL for November, 1927, p. 539-547, regarding the application of industrial principles.

It is Captain Thee's belief that the unit-replacement system can be applied in either a small or a large service-station, as well as by fleet operators, if sufficient thought is given to the subject and each case is analyzed sufficiently so that a satisfactory solution can be worked out. The unit-replacement system of maintenance of motor transportation has been used exclusively by the United States Army, he continued. Whenever it has been properly applied, he said, it not only has been very successful but very efficient and economical. The unit-replacement system is not being accepted by many of the large factory service-stations or repair shops of large-scale fleet-operators for the following reasons, he remarked.

(1) Managers in charge of maintenance of motor-vehicle fleets or large service-stations are not willing to accept and use the system

(2) It has not proved to be more economical. This is due to the fact that unit assemblies have not been built economically in mass quantities

(3) The equipment of large-scale fleet operators is not standardized

(4) Failure of manufacturers of motor-vehicles to cooperate with fleet operators and factory service-stations in regard to making arrangements to repair unserviceable units

Perhaps the ideal solution for the adoption of the unit-replacement system by fleet operators, said Captain Thee, is not to buy vehicles complete but to purchase units and build the vehicles with them. He quoted Col. Edgar S. Stayer, Q.M.C., Commanding Officer, Holabird Quartermaster Depot, in his paper on Standardization of Military Motor Transportation published in the March-April, 1930, issue of the *Quartermaster Review*, as follows:

The standardization upon a special type and make of vehicle, so far as military motor-transportation is concerned, is not considered as the proper method of standardization, but rather a standardization based on written specifications describing the units which will be used to build this transportation, and which are known by service and testing experience to be of sufficient durability and ability for the military needs.

In simplification of cargo transportation for the military service there appear to be three salient factors: (*a*) The number of different vehicles that is required; (*b*) the types and capacities that will be adopted; and (*c*) the interchangeability of units among the different types of vehicle.

Operation of the foregoing type of equipment will simplify repairs by the unit-replacement system, Captain Thee remarked.

Captain Thee also commented upon wastage and stock reserves, and considered accessory units. In conclusion, he stated that it is very doubtful whether manufacturers of motor-vehicles desire to adopt the unit-replacement system as this system for making repairs will, no doubt, increase the life of a motor-vehicle and therefore reduce sales. The principal factor which will determine when a vehicle should be discontinued from service is obsolescence, he said.

The Oral Discussion

Asked by H. W. Alden what efforts are made by the Public Service Coordinated Transport of New Jersey in the way of penalties against drivers for accidents and regarding the bonus and honor system for freedom from accidents and excessive maintenance cost of the vehicles they operate, Mr. Lyman replied that this company has a bonus system for freedom from accidents. They receive a bonus every month during which they do not have an accident, a conditional bonus every three months, and a yearly bonus; that is, if they drive for a year without an accident, they receive another additional bonus. A driver's record is kept very carefully. Each time an operator has an accident, it is recorded on his record card, which is watched very closely. If an operator has a number of accidents, he is reprimanded; if he has several accidents, he is laid off for a period; and after having had a certain number of accidents he is discharged.

Mr. Glynn said that the Bell telephone system is one of the foremost exponents of accident prevention and safety. It believes that there is only one way to do any single operation and that is the safe way; therefore, safety and accident prevention are a part of every man's regular job.

A. F. Coleman said that, while operating the Peerless petroleum field in the West, a strike occurred. During the strike this company closed its shops through a certain section of the West and operated on the basis of using manufacturers' repair shops as a means of taking care of the equipment.

Leo Huff remarked that, in the last 10 years of the experience he has had of operating fleet-equipment, he has operated under the various systems and has found that a man taking over the duties of operating a fleet of equipment, regardless of how small or large it may be, must study the individual fleet. Ralph Baggaley, Jr., stated that, in considering whether or not we will have self-service or service-station maintenance by the manufacturer, we must consider more than the cost per mile for the vehicles operated, because we also should take account of the number of days out of service that either of these services will give.

Temple C. Smith said that there is a very definite relation between the maintenance and the depreciation of motor-vehicles. It was mentioned in one paper, he said, that it might form a good incentive to have three or four different garages operating under the same general management so as to set up competition between them as to the maintenance cost on the vehicles. That really would not mean very much, he continued, because the man who is sharp will trade-in his cars earlier and keep down his maintenance, which, from an over-all viewpoint, may very possibly be the wrong thing to do from an economic basis. If low maintenance is desired, he remarked, trade in your cars early and do not run them through their probable economic life. Mr. Brede remarked that the general discussion seems to lead toward an agreement that a combination of the systems is still the right answer at this time. Mr. Marble agreed with Mr. Brede's statements and said he believed the whole question, especially with reference to the concentrated fleet, is open for individual study. This led M. C. Horine to say that in one case, service-station maintenance may be more expensive, and, in another case, cheaper. He believes that no adequate general answer can be made to a general allegation in this case.

Other subjects touched upon included the running repairs and maintenance of a fleet handled in the owner's garage, major repairs in the manufacturer's service station, repairs which can be made overnight, and similar topics.

In conclusion, Mr. Middleworth summed up the voluminous discussion and said that he is rather inclined to agree with what seems to be the majority opinion; that each individual fleet-owner should make a very careful survey of the conditions under which

he operates and then decide whether self-maintenance or service-station maintenance is the best method to pursue. He believes that it is not possible to make any set and fast rule as to how and when vehicles should be reconditioned, or to specify in detail what methods should be adopted.

Transmission Session a Forum

Three versus Four-Speed Passenger-Cars and Constant-Mesh or Sliding Gears Discussed

THE Transmission Session convened in the main convention hall at 8:30 p.m., Monday, May 26, with T. J. Litle, Jr., as chairman. Three papers were presented, the first being entitled Comment on American Passenger-Car Gearsets, by Herbert Chase. In his paper modern transmissions were discussed and criticized primarily with respect to ease of gearshifting, quietness and relative simplicity. The author drew no final conclusions but left the impression that most, if not all, alleged advantages of four-speed transmissions can be obtained with the simpler, less expensive, three-speed type that requires less gearshifting. He told also what has been achieved by three-speed advocates with respect to quiet operation, easy gearshifting and longer car-life without sacrifice in general performance. The paper is printed in full in this issue beginning on p. 727.

Passenger-Car Gearsets Discussed

In the discussion, C. E. Swensen, chief engineer of the Mechanics Universal-Joint Co., strongly urged the use of comparatively small axle-reductions as the best means of obtaining pleasant high-speed car-operation. This naturally calls for a high power-weight ratio and, he said, coupled with a gearset having a quiet, next-to-high or direct gear-ratio, constitutes a car that is much more satisfactory to drive than does one having conventional axle-ratios.

Mr. Swensen remarked that he drives cars of both types. If, after driving the fast-axle car, he immediately drives the low-gear job, he at once feels that the engine is racing. He thinks also that the maker of the fast-axle job seems to have succeeded in providing an engine that makes the power-weight ratio such that the amount of gear-changing is not noticeably greater than that to which he always has been accustomed.

New Type of Transmission Described

A. Bjorkman, of Stockholm, Sweden, research engineer for the Spontan Co., said, in part, that as the speeding-up of a car represents mechanical work performed by an engine of limited capacity, it is important, for rapid acceleration, that the engine should be run as close to its maximum output as is possible during the entire acceleration period to deliver the necessary work in the shortest possible time. This means a continuously varying gear-ratio, he continued. A four-speed transmission obviously gives the driver possibilities to keep the gear ratio closer to the ideal one than does the three-speed gearset, he remarked, but it depends on the driver whether the greater possibilities of the four-speed transmission in this respect are utilized by more frequent gearshifting.

His transmission, said Mr. Bjorkman, has neither clutch nor gears. Its operation depends upon centrifugal forces of two rotating weights acting on two over-running roller clutches. The use of centrifugal forces for transmitting the drive makes the action extremely smooth and flexible at all speeds. It is noiseless because of the absence of gears, and its efficiency is about 98 per cent.

Important as the technical advantages of increased acceleration are, Mr. Bjorkman continued, the main advantage of this transmission to the motoring public lies in the greatly simplified handling of the car because gearshifting is abolished. The transmission prevents the engine from becoming stalled, and the inherent features of coasting when the engine is not driving and an automatic "no-back" device are very useful in hilly country.

Types of Transmission Available

Four types of transmission are available from which to select—the spur, the herringbone, the helical and the internal types—said Carl Nearacher. In selecting the most suitable one, he continued, the major considerations are general excellence balanced against cost. Generally speaking, the cheapest is the conventional spur type, the most expensive is the internal type, and the herringbone and the helical types come in between. These cost comparisons are based on design only. To produce a very quiet transmission makes the cost a very different consideration in which a more expensive design may be the cheapest production proposition.

Mr. Nearacher said also that the last 18 months has been a stage of manufacturing development. "We feel," he said, "from our company's experience, that we have gained the knowledge required to produce transmissions of the internal type. In his opinion, the public is rapidly being weaned away from fast axles, or high engine-speeds at high car-speeds, and that the four-speed transmission has accomplished much of this."

Explanation of Cadillac Yoke Mechanism

C. V. Crockett quoted the following statement in the paper "The yoke mechanism includes a pair of oil plungers or dashpots arranged to prevent automatically excessively rapid shifting," and said that this is not wholly correct as the dashpots are not of sufficient power to prevent rapid shifting in case an owner desires to do so. They are, however, he continued, the connecting link between the shifting mechanism and the friction-clutch control-mechanism. It is the resistance of the dashpot which determines the power with which the friction clutches will be engaged. While this is seemingly a trivial point, the entire success of the transmission depends upon these dashpots.

If the gears are shifted slowly, said Mr. Crockett, little force is needed and, since the dashpot plunger will be depressed slowly, little force is applied. However, he remarked, upon a rapid shift, the dashpot presses the clutch into much harder engagement and applies the more powerful synchronizing force which is needed under these conditions. In a like manner, he said, resistance of cold oil in the dashpots assures the added pressure needed to synchronize gears rotating in cold oil and hence having a tendency to stop too quickly.

Constant Mesh or Sliding Gears?

The second paper was entitled Constant-Mesh or Sliding-Gear Transmission, by F. C. Pearson, of the Reo Motor Car Co. He said, in part, that one of the most important developments in the last year has been the production of better transmissions. Like all other of the major units of the car the transmission has had its day at one time or another during past years. When the three-speed sliding gears superseded the two-speed planetary transmission it was signaled as an engineering achievement. Several years later the four-speed transmission was given much publicity. With the advent of more cylinders and better engines transmissions returned to the conventional three-speed type until recently. We are now in another transmission era. This time developments point toward better, quieter and more useful transmissions whether built with three or four forward speeds. Road conditions have improved to such an extent in the last few years that cars are constantly being driven at higher average-speeds.

This speed greatly lessens the life of engines, making one of two things necessary; either building larger engines that would prove more expensive to operate, or gearing the cars so that high speeds can be obtained without too high engine-speed. The present tendency is to build transmissions that will have high-gear quietness in second or third gear, which makes possible the use of lower ratios when desirable.

With respect to the development of the herringbone type of gear, Mr. Pearson said that the advantages of this type over the internal gear are simplicity, lower cost and lower weight. As the same number of gears and bearings are used as in the conventional clash-type, the bugbear of complicated construction was at once removed. Gear thrusts are always present when helical gears are used but are not present in the herringbone construction, since each pair of gears absorb the driving thrust within themselves. This is the important point as balancing the end thrust arising when driving through two pairs of plain helical-gears is impossible. In this case thrust bearings have to be relied upon to take care of the unbalanced thrust-load and maintain the location of the gears. The multi-jaw clutches used in the three constant-mesh types of transmission seem to be very similar in design except for jaw shapes which have been worked out to fit their individual cases.

After many experiments, the speaker remarked that using two pitches in both constant mesh and second speed was found desirable. However, he said that the greatest improvement is shown when two pitches are used in the second-speed set and that the selection of the old pitch in the second-gear train was limited so that a final decision depended upon the well-known cut-and-try process rather than on mathematical fact.

Assembly is no more expensive or difficult than for the conventional transmission, said Mr. Pearson. Testing still presents the same difficulties regarding noise as in the spur-gear transmission. It is done in a quiet room at the end of the transmission-assembly line. Bearing loading is slightly greater for herringbone than for spur gears in transmissions where the center distances between main shaft and countershaft are the same. Transmissions of this type are driven probably 50 per cent more and much faster, in gear, than the conventional type. This requires more bearings to maintain the same life. Using anything other than standard steels regularly used by conventional transmission-makers has not been found necessary. The design of this transmission allows the use of bar stock and making all of the gear blanks in automatic machines. This is a saving over forged blanks and seems to give gears that are more uniform in grain structure and have less eccentricity after heat-treatment.

Replying to a question asked during the discussion, Mr. Pearson said that the different pitches are used only on the second-speed set. All who are acquainted with gear cutting know that to cut gears which have a ratio other than 1 to 1 requires a very peculiar set of cutters to cut two pitches and mate them together, he said. The 1 to 1 ratio is the only one that can be cut with standard cutters; any others would require special cutters.

CHAIRMAN AND SPEAKERS AT THE TRANSMISSION SESSION

(Upper Left) Past-President T. J. Litle, Jr., Chairman. (Upper Right) Herbert Chase, Who Gave a Paper on American Passenger-Car Gearsets. (Lower Left) F. C. Pearson, Who Presented His Paper on Constant-Mesh or Sliding-Gear Transmissions. (Lower Right) S. O. White, Who Read the Paper on Recent Transmission Developments

Recent Transmission Developments

In presenting his paper on the above subject, S. O. White, chief engineer of the Warner Gear Co., said in part that until the last few years passenger-car transmission-design followed a few well-beaten paths and consisted largely in trying to get the greatest capacity possible into the least space, using the lightest gears and most compact and economical arrangement consistent with reasonable service performance. Along with this there was constant progress in the manufacture and heat-treating of gear steels and in the gear-cutting art. Three forward speeds and a certain amount of gear noise were generally accepted in this Country, although there was always the feeling that there might be a way to decrease the noise.

Suddenly, said Mr. White, the quiet-running properties of internal gearing were called to our attention, and along with it came a revival of the four-speed

transmission but using the internal gear for the next-to-top speed. Four-speed transmissions for passenger-cars were nothing new and their advantages were known, but the noise stood in their way, the speaker continued. It was essential to provide a sure and easy shift between the two top-speeds, but the nature of the internal-gear design made this fairly easy. As a consequence, he said, several four-speed internal-geared transmissions have come on the market and have achieved noteworthy popularity.

Summing up three as against four speeds, said Mr. White, some advantage can be claimed for each. Unquestionably, slowing down the engine makes a sweeter running automobile; also, multiple speeds, properly used in individual chassis designs, can increase car-performance. Engineering opinion has said that the public does not want to shift gears. With the advent of easier gearshifting, this opinion perhaps should be modified. But, he said, after all is said and done, the public is going to render the decision.

Brakes Thoroughly Discussed

Electrically Operated Vehicle Brakes and Brake Design Studied at Tuesday Morning Session

TWO PAPERS were presented at the Brake Session held Tuesday morning, May 27. The subject chosen by A. W. Frehse, of the Chevrolet Motor Co., was the Fundamentals of Brake Design. John Whyte, of the Warner Electric Brake Corp., treated the subject of Electrically Operated Vehicle-Brakes. W. R. Strickland was chairman and a large number of representative members and guests were present.

Fundamentals of Brake Design

It was remarked by Mr. Frehse in the introduction of his paper that modern brake-development is now reaching a stage in which the performance of a brake design can be predetermined accurately with almost the same certainty as that for powerplants. Accurate mathematical expressions and formulas now supplant the old cut-and-try method used on many types of brake, he continued, and, armed with these mathematical tools, a designer can proceed with his designs without the usual misgivings and uncertainties that characterized former brake-design. In practically every case, the original layout using the principles outlined has withstood the rigors of breakdown testing and field service, with very few modifications.

Thermal and dynamic effects were disclosed in Mr. Frehse's paper, and their influence on the parts constituting the brake were discussed. These effects may spell success or failure in what looks to be a good design, he said. He also discussed the fallacies of the simple shoe-brake and offered a method for overcoming them.

Drum Warping and Wear

During the discussion, A. Y. Dodge, of the Bendix Aviation Corp., South Bend, Ind., called attention to the fact that drum warping is almost opposite to the natural movement of the shoe or what one might call the natural lining-wear diagram, and said that they tend greatly toward offsetting each other. After looking at thousands of shoes whose wear diagrams show that a great deal more wear should take place in the middle of the shoe than at the end, he remarked, we find that it is almost uniform if one takes an average of a large number of shoes; that is, when the drums are properly proportioned to the shoes. In other words, he continued, we have a shoe which changes its shape a little bit with heat, we have a drum which changes its shape with pressure, and those changes tend to counteract each other. We feel that we can make a brake which, with proper shoe-construction and proportion, will blend those factors together in a simple form that will accomplish much that Mr. Frehse pointed out as having been accomplished with the links.

Mr. Dodge suggested that the subject be confined to a simple design of shoe, self-actuating, with the drums revolving in the same direction as the shoe is revolving, comparing it with the simple-design shoe mounted with the links.

Effect of Braking on Tires

B. J. Lemon, of the United States Rubber Co., remarked that the point brought out that the brake must absorb five times the power at some particular period that the engine can produce in that same period is interesting to the tire industry. He asked Mr. Frehse whether, over a general running in traffic and on the road of a number of thousands of miles there is more power required to stop the car through the tires than to accelerate the car, keeping in mind that this ratio of 5 to 1 may be brought into play in a number of cases for deceleration. That is, are our tires being worn off more by deceleration than by acceleration, forgetting the regular run between the starting and stopping?

Mr. Frehse replied that it is rather difficult to make such measurements. The tires stand up fairly well, he said, in spite of the brakes. The acceleration force is much less than that of retardation. He stated that the best acceleration one gets with a car in high gear represents about 4 ft. per sec. per sec. Brakes, in some instances, will easily develop as high as 30 ft. per sec. per sec.; but, normally, about 20 ft. per sec. per sec. is the usual braking for quick stopping, driving up to the light and stopping in the last 10 or 15 ft. from 25 m.p.h. The tires seem to stand that test all right, he said.

Chairman W. R. Strickland said that he remembers examining shoes many times that would illustrate too much power at the rear wheels or too much braking; too much tread wear, for instance, too much side slippage and things of that kind. He thinks that if the braking were more severe it would have shown up in that way. "I do not mean slipping on the pavement, but one can generally see the pull of the rubber," he said. Consequently he thinks that, even with the increase in the braking, the natural wear and tear due to the driving still predominates.

W. C. Keys, of the United States Rubber Co., remarked that Mr. Lemon's question was confined to the difference in wear on tires in acceleration versus deceleration, neglecting driving conditions. He said that in any of the cities to which one is accustomed, one will see more rubber dust on the near side of the traffic light where people have to stop than one does on the far side where they are accelerating. In Mr. Key's opinion, there is a considerably greater wear from applying brakes than from accelerating.

Chairman Strickland concluded this discussion by remarking, "That brings up the question whether they are using these four-speed gearsets or not, and whether they have power enough. This will vary from year to year, depending on the ratio of power to brakes."

Electrically Operated Brakes

The main points covered by Mr. Whyte in his paper are as follows: The users of motor-vehicles are demanding more positive brake-control with the exertion of the minimum of physical effort. This is particularly true in the commercial field where large loads must be handled and comparatively high speeds are being obtained. There are definite limitations to the amount of self-energizing obtainable in existing brake-systems. An increase in brake output beyond that obtainable by accepted methods of self-energizing and with the use of reasonable pedal-pressures, is desirable. The

use of boosters is of assistance in this respect.

Continuing, Mr. Whyte said that electrical energy can be used as a means of energizing brakes, making it possible to obtain any predetermined brake output without effort on the part of the operator. The electrical system offers advantages due to the ease of electrical distribution and simplicity of installation.

Mr. Whyte stated the basic forces involved in braking, the limitations to available leverages, the gain due to use of self-energizing, and explained external pressure boosters. Further, he discussed equalization problems, the effect of variable friction and axle braking-ratios, as well as front-axle reactions.

As to energizing brakes electrically, Mr. Whyte said that the factors he had already mentioned and the factor of simplicity of installation led to the development of the electromagnetic brake. He then analyzed this type of brake, saying that a definite amount of kinetic energy is available in every moving vehicle, and that this energy must be absorbed to stop the vehicle. The electromagnetic brake simply utilizes the kinetic energy of the vehicle so that it can stop itself. The author then gave reasons why this brake is nonadjustable, commented on automatic reverse braking and brake performance, and said, in conclusion, that it is generally conceded that some method other than mere physical force on the part of the driver must be employed to obtain satisfactory brake performance with reasonable pedal pressures. To get the best results that are possible by the use of externally augmented brakes, whether of booster or of power type, the following salient points require consideration from chassis engineers:

(1) If satisfactory stopping of vehicles is to be accomplished, means must be evolved to increase the ability of front-end design to resist brake torsional forces

(2) These means should include precaution against reversal of caster

(3) They should contemplate the absorption of braking forces independent of the spring suspension. This is advisable to eliminate the irregularities arising from unequal deflection of opposite front springs. Some types of spring shackle also allow differences of position to take place in opposite ends of the front axle

(4) The positon of the front steering-arm ball should be worked out as closely as possible to a neutral position, so that the change in front-axle positon during brake applications will not impose any appreciable load on the steering-gear connecting-rod. If this is not done, the control of the front end of the car will be erratic and give the impression of "wandering."

Clearance Between Drum and Rim

In the discussion, and replying to questions asked by D. P. Forbes of the Gunite Corp., Rockford, Ill., Mr. Whyte said the prime factor in all high-duty braking is to get rid of the heat. In the passenger-car job it can ordinarily be done because the drums stay in the airstream under the car, he remarked. On motorcoach jobs, particularly with dual pneumatics and disc wheels, the drums are very carefully tucked-in out of the airstream, and it is the easiest thing in the world to cook pneumatic tires.

As to the relation between the coefficient of friction of the lining and the road, he does not think that one has very much to do with the other, Mr. Whyte continued. So far as braking efficiency is concerned, the coefficient of adhesion on the road is a widely varying factor. He has seen cases where the coefficient was apparently more than one, and that is because the problem really gets out of the friction class.

Cost Considerations

Asked by W. C. Keys about the cost of this type of brake, installed, as compared with other typical brakes, mechanical or possibly hydraulic, Mr. Whyte said that the comparative cost of a set of electrical brakes installed on the chassis does not exceed an average mechanical or hydraulic brake hook-up. Mr. Whyte said also that there is no clearance left between the armature and magnets ordinarily for normal running, and that it is simply a touching contact, without any lubrication. The friction drag is about 8 or 10 oz. on the tire at the maximum, he said.

The remainder of the discussion centered largely on further data and further details relating to the brake described by Mr. White. Drum distortion is well brought out in experiments on motor-truck brakes 4 or 5 in. wide, he said in conclusion. The brake liners showed that maximum wear is on the inner edge all the way around, due to the "coning" of the drum.

CHAIRMAN AND SPEAKERS AT THE BRAKE SESSION

Left to Right—A. W. Frehse, Who Presented a Paper on The Fundamentals of Brake Design; Past-President W. R. Strickland, Chairman; John Whyte, Who Presented a Paper on Electrically Operated Vehicle Brakes

Wind Resistance and Comfort

Effects of Body Shapes at Different Speeds—Physical and Mental Effects of Riding

THREE excellent papers delivered at the Body Session on Monday forenoon should go a long way in throwing light on the waste of engine power in overcoming air resistance at high car-speeds and in indicating how driving and riding produce fatigue as well as what persons riding in cars find objectionable.

Although the attendance at the session was not large, those who took part in it were keenly interested and remained after the close of the discussion to try the two wabblemeters brought to the meeting by Dr. F. A. Moss for demonstration purposes. The erratic graphic charts thus recorded on the No. 4 wabblemeter were not indicative of fatigue produced by sitting in the meeting, although the amount of this, if any, might have been ascertained had records of the same subjects been taken before and after the session.

The paper on Bodily Steadiness, an Index of Riding Comfort, which was given by Dr. Moss, was the third at the session and evoked the largest amount of discussion. It was a progress report on this phase of riding-qualities research directed by the Research Committee. The report in full, with illustrations, is printed in the Automotive Research section of this issue of THE JOURNAL.

Air Eddies Reduce Speed

A. E. Northup, of the Murray Corp. of America, presided over the session and lost no time in presenting, as first speaker, Felix W. Pawlowski, of the University of Michigan, who gave his paper on Wind Resistance of Automobiles in condensed form and illustrated it with numerous lantern slides. He pointed out that, as the rolling resistance of cars diminishes with the improvement of the highways, the air resistance becomes more apparent and at speeds around 40 m.p.h. begins to dominate. He gave three reasons for dealing with air resistance by aeronautics instead of hydrodynamics, although the latter science is the older. The three fundamental laws of aerodynamics were stated as follows:

(1) Resistance to motion is directly proportional to the density of the medium.

(2) Resistance is proportional to the square of the size of the body, or, more correctly, to the square of any homologous dimension in geometrically similar bodies of different sizes.

(3) Resistance is proportional to the square of the velocity of motion.

The simple mathematical symbols in which these laws are expressed were written on the blackboard by Professor Pawlowski, who said that the three laws can be combined in a proportionality that can be expressed in the equation $R=k\rho Av^2$, k being referred to as a coefficient of air reaction or air resistance, a quantity independent of the density of the medium, size of the body and velocity of translation but depending upon the shape of the body, its attitude relative to the direction of motion, viscosity of the medium and to some extent upon the smoothness of the body surface. Engineers prefer, he said, to write the equation simply $R=kAv^2$.

The total resistance to motion of a body immersed in the medium was subdivided by the speaker into (a) eddy-making resistance, (b) skin-frictional resistance and (c) induced drag, which appears only when the body, because of its shape or attitude relative to the direction of motion, generates a cross-wind force.

Disturbance Caused by Different Shapes

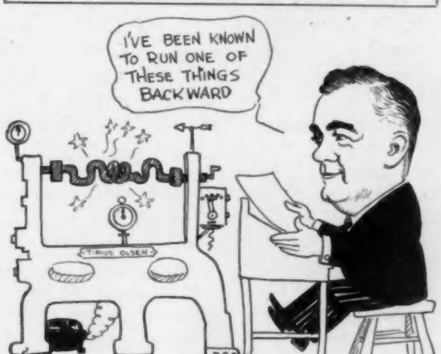
To illustrate visually the phenomena of air resistance, Professor Pawlowski showed slides of diagrams of the eddies behind a flat plate moved through the air in a position normal to its plane, larger eddies formed behind a plate concave to the direction of motion and the smaller eddies behind one convex to the direction of motion. He then showed flow lines around a parallelopiped having the same frontal area as the plate, around a sphere and, finally, around a streamlined body. He stated that the coefficient of resistance of the sphere is only one-seventh of that of the flat plate, while that of the streamlined body, around which the air flows smoothly with the least possible disturbance, is only about one-thirtieth of that of the flat plate.

Five different methods that have been used or suggested by different investigators for ascertaining the air resistance of automobiles were listed and the results of the tests made by them were summed up by the statement that they indicate that the coefficient k used in the engineers' equation lies somewhere between 0.0014 and 0.0018 lb. per sq. ft. per m.p.h. for closed cars and between 0.0016 and 0.0026 lb. per sq. ft. per m.p.h. for open cars of different standard types of automobiles in use at the time of the tests; that is, from 57 to 45 per cent and from 51 to 20 per cent respectively smaller than the coefficient of resistance of the flat plate. At a speed of 50 m.p.h. the wind resistance of the usual motor-car is between 110 and 170 lb. per sq. ft. and absorbs between 15 and 23 hp. At the strived-for speed of 100 m.p.h., these figures would be quadrupled.

Reflection, or Ground Effect

Next the speaker proceeded to show by diagram and to explain the principles and magnitude of aerodynamic reflection or ground effect caused by increase in air velocity underneath the body when it moves along the ground. This increase in velocity of air-flow results in decrease of pressures along these lines, which in turn produces a change in the magnitude and direction of the air reaction, that is, a cross-wind force that pushes the body toward the ground. From this the dictum is evolved that the body should be streamlined so as not to produce that force in proximity to the ground, which can be done by cambering the center line of the body.

What Has This To Do With Down Draft?



FRED M. ZEDER.

(N. B.—There is no intention to imply that the subject of the picture above is a taffy-twister. Ed.)

Referring to experiments made by a group of students at the University of Michigan in the year 1928-1929, with a series of cast-iron toy automobiles, of which slides of photographs were shown, Professor Pawlowski said that the tests were made at speeds of 60, 70, 80 and 90 m.p.h. These tests showed increases in resistance due to ground effect ranging from 21.2 per cent at 60 m.p.h. to 15.3 per cent at 90 m.p.h. for a phaeton with curtains up; from 14.5 per cent at 60 m.p.h. to 12.6 per cent at 90 m.p.h. for a phaeton with curtains down; from 3.0 per cent at 60 m.p.h. to 3.3 per cent at 90 m.p.h. for a coupé; and from —3.0 per cent at 60 m.p.h. to —5.5 per cent at 90 m.p.h. for a sedan. Therefore, the ground effect results in an increase in air resistance for phaetons and coupés and in a decrease for sedans.

How Rounding Corners Helps

Streamlining of all parts of the car exposed in motion would be necessary to effect the maximum reduction of eddy-making and air resistance. The

major portion of the eddy-making resistance, however, is due to the bulky body rather than to axles, wheels, fenders and lamps. Various examples of extreme streamlining were shown, and, for contrast, some early cars totally devoid of any attempt to reduce air resistance. Air-tunnel tests with models at the University showed, however, that the mere rounding of the edges and corners of a body on a radius of about one-sixth of the width of the front of a parallelopiped reduced the coefficient of resistance to less than 10 per cent at speeds from 60 to 90 m.p.h., as against a coefficient of 64 for a flat plate and of between 35 and 40 for a square-edged parallelopiped. Strangely, however, a much greater rounding of the edges and corners indicated that, instead of decreasing the resistance further, it decreased it less at these high speeds.

Reduction of Induced Drag

Considerable advantage was similarly shown to be obtained by beveling the flanks of the parallelopiped in substantially the amount of the conventional automobile, and the speaker also analyzed the aerodynamic design of the racing car with which H. O. D. Segrave established the straightaway record at Daytona that still stands. While even a slight improvement in the streamlining of that car cannot be imagined, he said, a more advantageous location of the leading and trailing edges could be suggested on the basis of consideration of induced drag and ground effect. Tests were made with models in which these edges could be disposed at seven different elevations ranging from the bottom level to the top level of the body height. These tests indicated that the ground effect increased the drag or resistance between 60 and 70 per cent for the different positions at 60 m.p.h. and by more than 100 per cent at 90 m.p.h. The zero-lift condition with ground effect was found to be with the leading and trailing edges at a height of 28 per cent of that of the body measured from the bottom of the body, and the condition of minimum drag agreed well with the condition of zero lift. With the correct location of the edges, induced drag is reduced from 38 per cent at 60 m.p.h. to 20 per cent at 90 m.p.h.

In conclusion, Professor Pawlowski remarked that streamlining of automobile bodies should be more extensively studied for the effect of lateral wind and that possibilties exist of reducing the air resistance of imperfectly streamlined bodies by artificially forcing the air to hug the body more closely; also, that the wind-tunnel scale-model method of studying the effect of some changes in design is most advantageous, since a great variety of modifications can be studied as to their comparative value at much less expense than could be done by experimentation with full-size cars.

How Cars Affect Riders

Prof. Ammon Swope, of Purdue University, was heartily applauded following his presentation of the second paper, on the Psychology of Riding-Qualities, prepared jointly by Dr. G. C. Brandenburg, of the University, and himself. He did not read the paper but gave it in the form of an explanatory talk as the lantern slides were shown. These showed in graphic form the data obtained from an investigation of the reactions of 125 subjects to rides in automobiles with the arithmetical mean or average indicated. The series included 27 charts covering 24 items on a list that the subjects were asked to check as to their preference. The items were classified as to the following groups of sensory qualities: motion, sound, sight, smell, spatial relations and esthetics. The subjects included 79 men and 46 women, about two-thirds of whom were university students.

The items on the list included speed, vertical motion, swaying motion, skidding, acceleration, deceleration, roar from car, body squeaks, noise from engine, noise from brakes, noise from tires, noise from wind, desirable amount of visibility, odors from engine, position of passenger in seat, kind of upholstery, arm and foot rests, depth of seat from front to back, height of front seat from floor, leg room, type of car body and kind of floor covering in front and in rear.

Twenty-four makes of car were listed in the study and the number of times they were mentioned ranged from 1 to 24.

Conclusions Derived from Data

The data obtained from the 125 subjects were tabulated by arithmetical means, standard deviations and probable error of difference of means, given in percentages, for each of the 24 items. The charts showed the degree of feeling on any item and the number of persons who rated at any given point of intensity of feeling or opinion.

Lack of time and space forbids going into details here of the information shown, and as the paper probably will be printed in full in a subsequent issue of THE JOURNAL, only the conclusions, which are tentative, will be summarized. On the basis of the data presented in the tables accompanying the paper, the author feels warranted in stating that

(1) Vertical motion, swaying and skidding are undesirable, the first being the most annoying

(2) Noises from various sources are very annoying, particularly body squeaks

(3) As wide a range of vision as possible is highly desirable

(4) The position and freedom of movement of the body of the passenger is very important

(5) The design of the seat is an important factor

(6) There are significant sex differences with regard to swaying motion, skidding, acceleration, deceleration, noise from tires and depth of seat from front to back

(7) The sedan type of body is almost universally preferred

(8) A definite preference exists with regard to floor covering, both front and rear

(9) The majority of men and more than half the women prefer to drive while riding

(10) Noise from any source is objectionable

(11) The comfortable qualities in riding are often mental, and fatigue may be due to these mental qualities as well as to bodily discomfort

The foregoing conclusions are felt to be little more than faint indications, as they are based on too limited a number of cases and the rating scale used is too crude, but it is believed that with further refinement the method will produce some valid and useful data. An instrument has been partly designed for measuring the effects of various motions on visual acuity and this is expected to be of considerable help in checking up on subjective judgments. The authors have also designed and built a steadiness-testing device that they hope to use as a further check. The next step is felt to be the application of the methods to several hundred subjects under the actual riding conditions.

Riding-Quality Research Valuable

When Chairman Northup called for discussion, W. C. Keys, of the United States Rubber Co., asked a number of questions regarding the 125 subjects, such as their age, prosperity, miles they had driven, whether they own the cars they drive in and the price class of the cars they drive or ride in, all of which are significant. Some of the questions were answered in the written paper but the answers to others were unknown. After Mr. Swope had answered as many as he could, Mr. Keys complimented him upon all the data he had presented and remarked that he thought they were of great value to the industry.

Dr. Moss then presented his paper on Bodily Steadiness, concluding with the statement that he wanted to emphasize mainly at this time the measurement of steadiness as one of the several factors in determining automobile comfort.

Mr. Keys then took the chair, as Mr. Northup had to leave to catch a train. He said that the work that Dr. Moss has been doing is fundamental and very valuable, and he asked to hear from President Warner, who had done considerable work on riding-qualities. Mr. Warner stated that Dr. Moss's researches are of absolutely fundamental importance and that those who have

CHAIRMAN, SPEAKERS AND DISCUSSERS AT THE BODY SESSION

Top, Left to Right—Prof. F. W. Pawlowski, Who Presented a Paper on Wind Resistance; A. E. Northup, Chairman; Dr. F. A. Moss, Who Delivered a Progress Report on Bodily Steadiness as an Index of Riding Comfort
Bottom, Left to Right—Tore Franzen, Prof. H. M. Jacklin and R. W. Brown, Who Discussed Riding-Qualities

been active in promoting the work of the Subcommittee on Riding Comfort have been surprised and disappointed that so little active interest in it has been displayed by the engineers who have not been directly engaged in the work of that group and on the part of the industry as a whole. The development of the means of measuring riding comfort so as to get away from the subjective determination and individual impression of those who take a ride in a car, is only the development of a means to an end in an ambitious program set forth at the Annual Meeting last January. He said he hopes that every member who has heard one of the progress reports will make himself a missionary to spread the news of the importance of making it possible to carry on the work on a liberal scale and with assured support for a number of years to come, as it should go on for an indefinite period. The results, he asserted, are going to become more and more interesting for at least two years and a larger number of papers on the subject will be presented if the industry rallies to the support of the riding-comfort group and the men who are carrying on the investigations.

Industry Can Apply Results

B. J. Lemon, of the United States Rubber Co., seconded President Warner's remarks, saying that 10 or 12 interested companies made contributions toward it last year and that other companies should be just as interested in it. In a year or two the industry can apply the results in a way that will be of real benefit.

W. S. James, of the Studebaker Corp., suggested that if as soon as possible drawings of a close approximation of what Dr. Moss thinks would be a suitable machine for the work were prepared and distributed to a number of car manufacturers so that they could build the machines and use them themselves, both the work and the collection of funds to carry it on would be furthered. He said he feels that any car manufacturer would find it of definite help if he had some more certain basis than personal opinion regarding the desirable riding-qualities of a car. He believes that progress along this line will be pushed ahead more rapidly if the research machines could be made widely available.

The suggestion made by Mr. James was a very sensible one, remarked R. W. Brown, of the Firestone Tire & Rubber Co., who has done considerable work on accelerometers for investigating car vibrations. His experience has been that the instruments developed more or less as a by-product of the main program in a project of this nature are appreciably ahead of the industry at this time and that they can be used in a practical way.

C. B. Veal, manager of the Research Department of the Society, stated that to him the remarkable feature about Dr. Moss's work has been the consistency of the results obtained with crude apparatus and the apparent certainty with which he has hit upon the wabblemeter as an instrument for measuring riding-quality.

Replying to an inquiry by Lowell H. Brown, of the Biflex Corp. and the Weymann Corp., as to just what the financing problem is, Tore Franzen said that it is necesary to raise $5,000 for this year's research by Dr. Moss, who has generously donated his own time. The Council of the Society has backed the project and will see that the work goes on but does not expect to drain the funds of the Society by making appropriations for it.

The discussion of Dr. Moss's paper was concluded by Merritt L. Fox, of the Houde Engineering Corp., who showed several lantern slides of gyro-accelerometer records taken from a 1925 and a 1930 model car of the same make at 10, 20, 30 and 35 m.p.h. showing the angular velocity of the front end when the wheels pass over a bump. These showed remarkable improvement in the riding-qualities of the new car over the old model and a large reduction of the number of vertical accelerations exceeding 19 ft. per sec. as recorded by a Firestone accelerometer. The records were made from the same two cars that the students used as subjects by Dr. Moss in his work rode in, so Mr. Fox had an exceptional opportunity to correlate and check the instrumentation with the results Dr. Moss obtained.

Streamlined Car Need Not Be Long

Referring to Professor Pawlowski's paper, President Warner pointed out that a method of measuring wind resistance that has been used with success in England is the drawbar pull measurement on a towed car.

Regarding the practicability of streamlining, he said that those who are familiar with the metalclad airship that has been expounded several times at S.A.E. meetings will recall that its over-all length is three times its diameter. It is possible to build an automobile with the conventional cross-sectional dimensions and no more than the conventional length and with very little elongation of the tail that will approximate the ideal streamline form.

Concerning the Segrave racing car, Mr. Warner said that he suspects that an accurate determination of ground effect on a car moving over the ground, which is impossible to get in a wind-tunnel, would show that the form of chassis cover that Segrave used was considerably better than was shown by Professor Pawlowski's measurement, indicating that an apparent saving of 20 per cent could have been made by raising the leading and trailing edges. Another difficulty in this respect with racing cars is that of keeping them on the ground at high speed and preventing them from rising in the air and becoming airplanes.

Horton the Golf Champion

Takes S.A.E. Men's Tournament Cup for 1930—Mrs. Funk Wins Ladies' Tournament

IDEAL weather throughout the week of the meeting brought out an unusually large number of golf players each afternoon. The courses were dry and in good condition, quite unlike their state at the Summer Meeting in 1927, when parts of them were under a foot or more of flood water, as many will remember.

The men's and the women's tournaments ended on Wednesday, with W. M. Horton winner of the men's tournament with a total gross score of 155. He turned in cards of 81 for the first and 74 for the final round, thereby winning the S.A.E. Member's Championship Golf Cup which he will hold until the Summer Meeting next year.

A. W. Anderson was the runner-up, with a total score of 157. He scored 81 and 76 for the first and the final rounds. He was awarded a handsome silver pitcher, the prizes being given to the winner's Thursday evening.

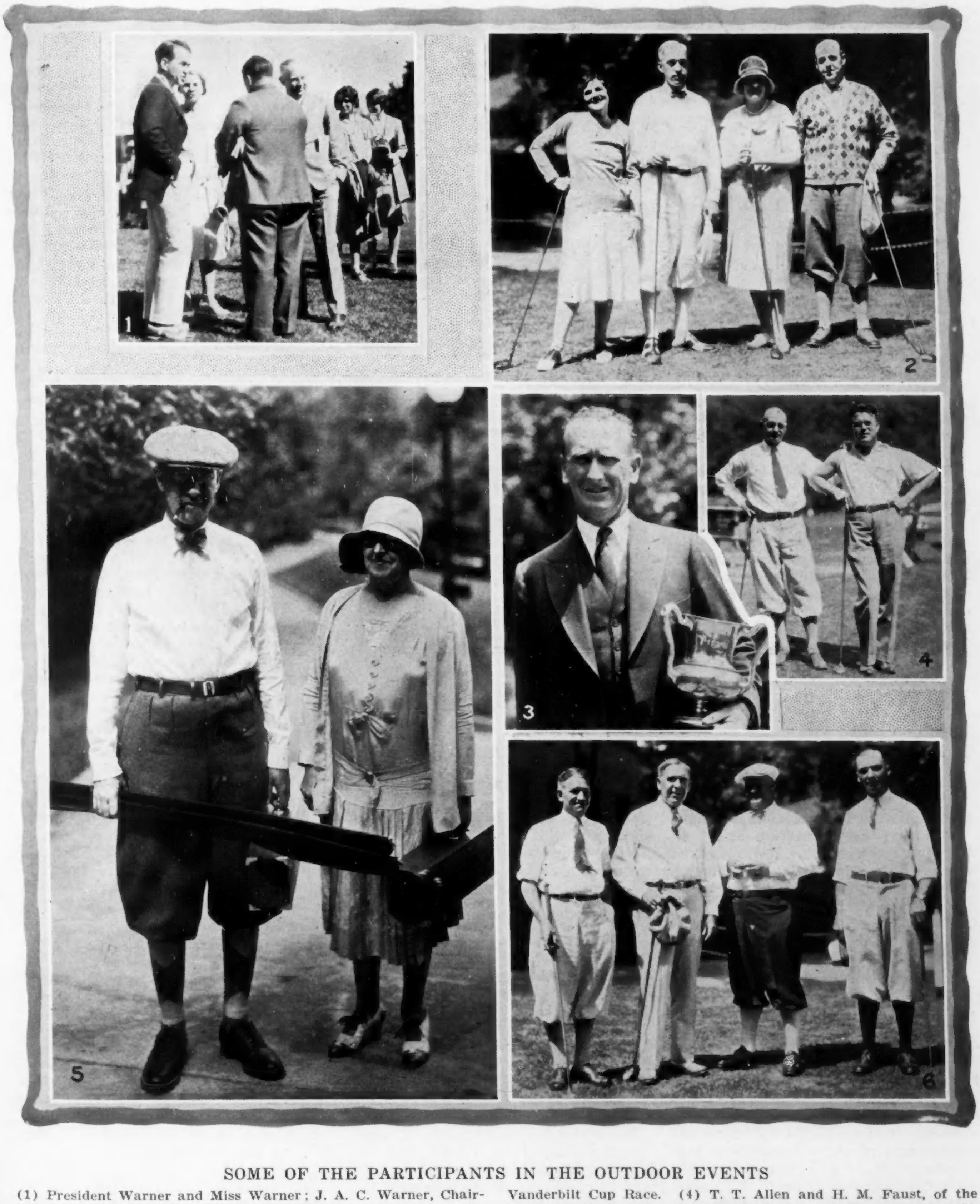
H. C. Tillotson and J. W. White tied, with net scores of 140 each in the A flight, but as Mr. Tillotson departed for home before the end of the tournament, Mr. White won this flight by default, with Mr. Tillotson runner-up.

The B flight was won by H. W. Kizer, with a total net score of 125. O. A. Parker and W. Fairhurst tied for second place, with a total net score of 135. The winner was decided by the toss of a coin, Mr. Parker winning.

D. G. Proudfoot and H. L. Ames played a tie in the C flight, with total net scores of 133 each. The toss of a coin gave the flight to Mr. Ames with Mr. Proudfoot as runner up.

Mrs. McCray Wins Handicap Flight

Mrs. J. B. Funk won the championship flight in the Ladies' Golf Tournament with a low gross score of 179 for 36 holes, shooting 90 in the first round and 89 in the final. Mrs. Stanley Whitworth was runner-up, with cards of 89 and 97.



SOME OF THE PARTICIPANTS IN THE OUTDOOR EVENTS

(1) President Warner and Miss Warner; J. A. C. Warner, Chairman of the Meetings Committee, and Dr. George W. Lewis. (2) Roy Frantz and Mrs. Frantz (Runner-up in the Women's Handicap Golf Tournament) and Their Guests from Milwaukee. (3) J. H. McDuffee, Chairman of the 1930 Golf Committee, with the Vanderbilt Cup Which He Won in a Steam Car in the First Vanderbilt Cup Race. (4) T. T. Allen and H. M. Faust, of the Curtis Publishing Co., Contestants in the Golf Tournament. (5) L. A. Chaminade and Mrs. Chaminade, an Enthusiastic Archery Couple. (6) H. E. Figgie (Left), Runner-up in the 1930 Golf Championship. F. W. Anderson (Right), Suggested Chairman of the 1931 Tournament

THE DETROIT SECTION HIGH-HATS
Who Provided the Music for Dancing in the Hotel Lobby Tuesday Evening

There were so many ladies in the tournament that the committee decided to have a handicap flight, with the understanding that the championship flight scores on the gross basis would take precedence. Mrs. H. D. McCray won this handicap flight with a low net of 147 for 36 holes. Mrs. J. R. Frantz was second, with a net score of 151 for her two rounds. Mrs. Whitworth had a low net of 138, but because she was runner-up in the championship flight, her net score could not be considered.

Joseph McDuffee, the Chairman of the Golf Tournament Committee for 1930, recommends that hereafter in tournament play all scores be made in foursomes.

Research Programs Considered

Progress and Future Work Discussed at Meetings of Research Committee and Subcommittees

RIDING comfort, front-wheel alignment, fuels, detonation and highway research projects were reported upon and plans for the next six months were outlined at a full schedule of luncheon and dinner meetings for members of the Research Committee and its Subcommittees during the Summer Meeting.

Following the Body Session on Monday, at which Dr. F. A. Moss presented a progress report on the fatigue research which he is conducting with the guidance of the Riding-Comfort Research Subcommittee, a luncheon meeting of that Committee was held. The wablemeters developed in the course of this work have been giving remarkably consistent results in indicating the fatigue acquired by automobile riding, and members of the Committee are enthusiastic over the possibility of wide industrial application of the instrument as a means for measuring the fatigue of workers engaged in routine production tasks. At least one such application has been made with success, affording a scientific basis for determining the most economic method of production from the standpoint of conserving human energy.

To Furnish Wabblemeter Drawings

To facilitate progress in this direction, R. W. Brown, Chairman of the Committee, volunteered to make detailed drawings of the wabblemeter which has proved most satisfactory in the work thus far, and the Committee agreed to make such drawings available to persons interested in making industrial application of the instrument.

The Subcommittee on Front-Wheel Alignment gathered for a dinner meeting on Monday night, but since the time was insufficient for covering the business in hand, the group reconvened at breakfast on Wednesday.

The scope of the alignment work carried on by this Committee was reviewed, including the variations in the specifications for caster, camber and toe-in as supplied to the Committee by the car maker and as actually found by measuring new cars. As many as 50 per cent of new cars measured did not conform to the car makers' specifications, while a considerable number met the specifications only because of a wide tolerance set up by them.

To Report on Alignment Instruments

It was felt that, to obtain conditions of good driving and free-rolling front wheels to avoid undue tire wear, rather close alignment limitations must be maintained by the individual manufacturer, and that competition would compel the car designer to provide and his service divisions to maintain such limitations.

Considerable discussion followed regarding instruments for measuring alignment both at the factory and in the service station, and a set of conclusions based on the results of the Proving Ground tests were drawn up and approved by the Committee. It was also taken as the sense of the meeting that the findings should form the basis of a paper for presentation at a forthcoming meeting of the Society.

Diesel Fuel-Oil Research

Since the annual meeting the Fuels Subcommittee has considered further the proposed programs of Diesel fuel-oil research and adopted the program substantially as outlined at that time by Dr. H. C. Dickinson, of the Bureau of Standards. The Committee also named Dr. Dickinson as guiding director in carrying out this research, with authority to seek cooperation with other laboratories that may be available and willing to undertake part of the work. Plans for financing the project were discussed at length and a decision reached that the Diesel-engine manufacturers and representatives of the petroleum industry should be approached by the Ways and Means Committee with a view toward securing sufficient funds for a three-year program.

Road-Impact Instruments Discussed

The Bureau of Public Roads has just issued a report covering the investigation conducted in cooperation with the Bureau of Standards to determine the accuracy and other characteristics of the instruments that have been used, or proposed for use, in the cooperative motor-truck impact investigation conducted by the Bureau of Public Roads, with the Rubber Association of America and the Society acting in a technical advisory capacity. The Highways Research Subcommittee considered the report at a luncheon meeting, and all members present were urged to attend the meeting of the Cooperative Committee on Motor-Truck Im-

pact Tests, which had been called by T. H. MacDonald, Chief of the Bureau of Public Roads, for June 3 in the City of Washington.

To Extend Vapor-Lock Investigation

The meeting of the main Research Committee at luncheon on Thursday closed the series of Committee meetings and considered reports from all the Subcommittees. At this session the actions taken by the Subcommittees, as herein reported, were indorsed by the Research Committee, and the projects under the direction of the Cooperative Fuel Research Steering Committee were reported.

The paper, Effect of Weathering in the Tank on the Vapor-Locking Tendency of Gasoline, by Dr. O. C. Bridgeman and Miss E. W. Aldrich; and the paper entitled Engine Acceleration, by C. S. Bruce, both of which were presented at the Research Session, as reported elsewhere in this issue; and also a paper presented by Dr. Bridgeman at the Aircraft and Aircraft Engines Session, entitled The Effect of Airplane Fuel-Line Design on Vapor Lock, were referred to as constituting progress reports of the various phases of the work being conducted at the Bureau of Standards under the direction of the Steering Committee. In addition, considerable discussion was held on the future program for extending the vapor-lock investigation into the automobile field, the plan including a cooperative program for securing fuel-line temperatures of various makes of car on the road. It was generally agreed that an outline of procedure, taking cognizance of the suggestions made at the meeting, should be drawn up and circulated to the companies that had expressed willingness to assist in the undertaking.

The Research Committee also indorsed the steps taken by the Subcommittee on Methods of Measuring Detonation of the Cooperative Fuel Research Steering Committee toward establishing a tentative standard reference fuel for antiknock work, and heard with interest the Detonation Committee's plans for reporting, at a symposium session at the next Annual Meeting of the Society, the research on the effect of various engine variables on knock-testing results.

Ladies' Bridge Tournament

WHILE the technical sessions were in progress in the forenoons, many of the ladies attending the meeting passed the time enjoyably in the bridge tournament on the mezzanine floor of the hotel. Most of the attractive first and second prizes were of a utilitarian domestic nature, as will be seen from the following record of winners and their scores:

	Winners	Score	Prize
	Monday Forenoon		
1st	Mrs. J. B. Funk	1933	Waffle iron
2nd	Mrs. L. I. Gibbons	1793	Pen
	Monday Afternoon		
1st	Mrs. J. S. Hughel	2068	Silver pitcher
2nd	Mrs. H. C. Dishman	1952	Bridge set
	Tuesday Forenoon		
1st	Mrs. D. B. Grasett	2311	Sherbet glasses
2nd	Mrs. F. W. Wright	1849	Clock
	Wednesday Forenoon		
1st	Mrs. H. A. Hansen	1819	Pitcher
2nd	Mrs. Ruth Body	1800	Tea set
	Thursday Forenoon		
1st	Mrs. F. M. Germane	1943	Tea set
2nd	Mrs. W. S. Howard	1727	Gold candy dish

Council Elects Clarkson an Honorary Member

SESSIONS of the Council were held at French Lick Springs on May 25 and 28, and were attended by President Warner, Past-Presidents Strickland and Wall, Vice-Presidents Davis, Horner and Scaife, Councilors Glynn, Herrington, Moyer, Parker and Teetor, Treasurer Whittelsey and Chairman Boor of the Standards Committee.

A financial statement as of April 30, 1930, was submitted. This showed a net balance of assets over liabilities of $247,025.29, this being $34,069.36 more than the corresponding figure for the same day of 1929. The gross income of the Society for the first seven months of the present fiscal year amounted to $251,535.10, the operating expense being $234,274.67. The income for the month of April was $34,581.05 and the operating expense during the same month was $29,607.63.

One hundred and four applications for individual membership were approved, together with 4 transfers in grade of membership and 2 reinstatements. Five applications were reapproved. Eighty-four additional applications, 12 transfers, 1 reinstatement and 3 reapprovals, which had been acted on by mail vote, were confirmed.

The Council voted that, should the Constitutional amendment regarding Student Enrollment, which is now under discussion, be approved, the Student Enrollment fee shall be changed to $3 a year for the college term and $5 for the year thereafter.

The Council approved the action taken by the Standards Committee at its meeting held at French Lick Springs on May 25, as submitted.

A report was made to the Council that Coker F. Clarkson had been duly nominated for Honorary Membership in the Society by 10 members and the necessary vote of the council had been duly taken; therefore Mr. Clarkson was elected by unanimous consent of the entire Council and with gratification that the first Honorary Membership in the Society could be bestowed upon a man who was so deserving of this tribute.

Indianapolis Race Information

Sturm Gives Philadelphia Section Meeting Details of the Entrants in the Memorial Day Race

MEETING at the Philadelphia Auto Trade Association rooms on May 5, the Philadelphia Section heard from Chairman Dalton Risley the formal announcement of the Section officers elected for next year, who are: Edmund B. Neil, Chairman; W. Laurence LePage, Vice-Chairman; L. E. Lighton, Treasurer, and J. P. Stewart, Secretary. The subject of the meeting was timely, because of an important race held at Langhorne May 3, and a number of prominent racing men attended it.

William F. Sturm, a well-known writer on racing and American manager for Major Segrave, Frank Lockhart and Kaye Don, gave a paper on the Indianapolis Speedway and the eighteenth annual International Sweepstakes, to be run on Memorial Day. He said that the track was originally paved with tar-bound crushed stone in 1909, but this surface proved so unsatisfactory that it was repaved with brick at the end of the same year. The course is 2½ miles, with 50-ft. straightaways of 3301 ft. at the front and back and 640 ft. at the ends. The four turns are each 1320 ft. long and 60 ft. wide, banked at 16 deg. 40 min. except for the outer 10 ft., which stand at an angle of 36 deg. 40 min. The approaches to the corners are banked 2 deg.

New Rules for 1930 Race

The first 500-mile race was held on Memorial Day in 1911, and the highest speed for the 500 miles was made in 1925, by Peter DePaolo. Rules for the 1930 race change the limit for piston displacement from 91½ cu. in. to 366 cu. in. Poppet-valve engines are limited to two valves per cylinder, carbureters are limited to one dual or two single, and the centrifugal type of supercharger is ruled out. The car bodies must have a minimum width of 31 in. at the cockpit; and two men must ride, for the first time since 1922. The car must weigh at least 1750 lb. and not less than 7½ lb. for each cubic inch of piston displacement.

Hopes were entertained that these rules would usher in a reign of stock-car racing; but most of the cars entered are highly specialized racing cars, according to Mr. Sturm, and many of them have seen racing service in previous years. Some of the 122-cu. in. cars, laid aside when the 91½-in. limitation was imposed in 1926, have been increased in bore and entered.

DATA ON CARS ENTERED IN INDIANAPOLIS RACE

Driver	Entrant	Car	Engine	Piston Displacement	Drive[3]
		Four-Cylinder Engines			
L. Allen	Leslie Allen		Miller[1]	183	
W. Cantlon	William White	Miller Hi-Speed		183	
A. Gulotta	James Talbot, Jr.	Mavv Carbureter Special	Miller Marine	151	
K. Kenealy	James Talbot, Jr.	Mavv Carbureter Special	Miller Marine	151	
C. Miller	T. J. Mulligan	Fronty-Ford Special		176[2]	
L. Moore	Coleman Motors Corp.	Coleman		183	Front
P. Shafer	Coleman Motors Corp.	Coleman		183	Front
W. Shaw	Empire State Motors	Empire State Special	Miller	183	
E. Triplett	A. Guiberson	Guiberson Special	Miller[1]	183[1]	
		Hoosier Pete	Clemons	197[2]	Front[1]
		Six-Cylinder Engines			
H. Butcher	Butcher Bros.	Butcher Bros. Special		331	
J. Gandino	Jean Gandino	Chrysler Special		288	
G. MacKenzie	Auto Engineering & Machine Co.	Ambler Special	Buick[1]	255[2]	
J. Slade	Julius C. Slade	Slade Special		288	
		Eight-Cylinder Engines			
C. Burton	Ira Vail	V-Eight	Oakland	251	
L. Corum[1]	Milton Jones	Stutz Special		322[2]	
L. Cucinotta	Letterio P. Cucinotta	Maserati Special			
R. Decker	Bessie Decker	Decker Special	Duesenberg	300	
W. Denver	Gabriel Nardi	Nardi Special	Duesenberg[1]		Rear
R. DePalma[1]		Duesenberg Special		244	
P. DePaolo	Peter DePaolo	Duesenberg Special		244	
D. Evans	David Evans				Front
F. Fansin	Fred M. Fansin	Fansin Junior Special			
F. Farmer	M. A. Yagle	Betholine Special		100	
C. Gardner	James H. Booth	Buckeve Special	Duesenberg[1]		
W. Gardner	W. H. Gardner	(Engine may be changed to a four)			Front
S. Greco	Samuel Greco	Scranton Special			
W. Arnold[3]	Harry Hartz	Miller		151[2]	Front
J. Huff	Herman N. Gauss		Marmon[1]	100[2]	Front
Litz	Henry Maly	Duesenberg Special		150[2]	
J. MacDonald	William H. Richards	Romthe Special	Studebaker[1]	377[2]	
C. Marshall	George A. Henry		Duesenberg	262[2]	
E. Meyer	Ezekiel Meyer		Miller		
C. Moran, Jr.	Du Pont Motors	Du Pont Special			
L. Schneider	Louis Schneider		Miller	147	
J Seymour	Herman N. Gauss		Marmon[1]	100[2]	Front
R. Snoberger	Russell Snoberger	Russell	Studebaker[1]	337[1]	
E. Stann	August Duesenberg	Duesenberg		142	
M. Trexler	Marvil Lain	Trexler Special	Auburn	298	
	William Alberti	Duesenberg Special			
	August Duesenberg	Duesenberg		150	
		Sixteen-Cylinder Engines			
B. Borzacchini	Alfieri Maserati	Maserati		244	
L. Meyer	Alden Sampson II	Sampson Special		201	Rear

[1] Reported but not confirmed.

[2] Data added from a published list.

[3] All cars not otherwise designated are assumed to be rear drive.

Several new 183-cu. in. four-cylinder Miller engines will be used.

A 16-Cylinder Racing Oddity

One of the most specialized cars entered in the race is the Sampson Special, which Mr. Sturm described in some detail. Its sixteen 2 5/16 x 3-in. cylinders are mounted in two side-by-side banks of eight, with separate crankshafts geared at the forward end to a central shaft on which the flywheel and clutch are mounted. The transmission and rear-axle drive are conventional. The two engine units are entirely separate, even having independent water circulation through a radiator which is divided vertically in the middle. The separation begins where the gasoline is fed to the two carbureters from a common gasoline tank of 38 gal. capacity, mounted over the rear axle. The wheelbase is 103 in. and the tread standard.

The car will weigh approximately 1950 lb., empty, and 32 x 6.00-in. racing cars will be used. It cost about $16,500, which Mr. Sturm believes to be the highest cost of any car ever entered in an Indianapolis Speedway race, in spite of previous reports of higher costs. Louis Meyer and Alden Sampson II have had a very successful partnership in racing since 1928, when Meyer won the Indianapolis race with a car which, Mr. Sturm said, had been begging for a purchaser. A measure of their success has been due to Riley Brett, the mechanical genius of the trio.

Along with all these special cars, several cars are entered which are reported by Mr. Sturm to have modified stock engines of various makes. Two of these, entered by Peter DePaolo, are rebuilt Duesenberg passenger-car engines, of a model current several years ago with special heads and others are of various popular makes, as indicated in the tabulation herewith. These two Duesenbergs have 114-in. wheelbases, longer than most racing cars, but considered just right by DePaolo.

Front-Drive Cars Show Speed

Of much interest in connection with present front-drive activities are the figures which Mr. Sturm gave for qualifying times for the races during the last several years. Each entrant must demonstrate a speed of 85 m.p.h. for 10 miles on the track before starting the race. The driver's position at the start of the race depends upon his qualifying speed, the highest speed being granted pole position in the first row of three cars. In 1927 two front-drive cars won positions among the first three, although Lockhart had the pole with a rear-drive car. In the two races since, front-drive cars have monopolized the front rank at the start. The highest qualifying speed in 1926, the first year of the 91½-cu. in. engines, was under 112 m.p.h., and the lowest speed among the first three qualifiers for each race since, with the exception of one car, was over 119 m.p.h.

With superchargers eliminated, Mr. Sturm's prediction was that the race would be won at 93 to 95 m.p.h., these figures being based on qualification speed made in 1922, the last year of the two-man cars. Chairman Risley said that one of the important reasons for the ruling out of superchargers was that those used on racing cars have been too expensive to keep running.

Several race drivers and mechanicians were present at the meeting, among them Joseph Dawson, James Gleason, James Hill, Wesley Crawford and Fred Frame. Mr. Dawson and Mr. Frame contributed to the discussion.

Mr. Sturm was questioned in regard to the experiences of Louis Coatalen and Kaye Don at Daytona Beach. He said that the American Automobile Association report stated that the beach was not in suitable condition for speed sufficient to break the existing record at any time while Mr. Don was there. In addition, considerable minor mechanical trouble was experienced. Much of this Mr. Sturm attributed to the late substitution of a centrifugal supercharger for the Roots-type blower with which Mr. Don had been familiar.

Kaye Don has no yellow streak, according to Mr. Sturm, and any stories of dissension between him and Mr. Coatalen are based on nothing more than ordinary differences of opinion which may have been magnified by reporters who were hard up for copy. Mr. Sturm said that the Silver Bullet probably did not ride well, as might be expected from its limited spring motion, but he reported Mr. Don as saying that it is, nevertheless, a wonderful car to handle.

Racing Topics Summarized

Indiana Section Debates Merits of Present and Former Indianapolis-Speedway Practices

IMMEDIATELY previous to the introduction of the speakers at the meeting held May 15 in the Chateau room of the Claypool Hotel, the following officers of the Section were unanimously elected as follows: Louis Schwitzer, chairman; Bert Dingley, vice-chairman; C. A. Trask, treasurer; and Harlow Hyde, secretary. A dinner was held and 90 members and guests were present. At the opening of the technical session, 240 members and guests were in attendance.

Following the opening of the technical session, T. E. Myers read his prepared paper on the preceding races at the Indianapolis Speedway and on those which are to be held on May 30. He said in part that with the race this year the Speedway celebrates its twenty-first birthday. Continuing, he remarked that this racing event was born as an engineering experimental station. Perturbed because American motor-car manufacturers had no suitable working laboratory in which they might test their conceptions, he said that Carl G. Fisher conceived the idea of a 2½-mile race track.

Mr. Myers gave a most interesting description of the history of development of this track and of the engines and cars which have operated upon it, as well as commenting upon high-speed racing cars and engine progress.

Past-President William G. Wall, a member of the American Automobile Association Contest Board, was unable to be present but sent prepared discussion which was read at the meeting. Mr. Wall agrees with Mr. Myer on what he says about the Speedway and its influence on racing. "I have just been thinking of what the lessons the Speedway races have taught us in the past and what they will teach us in the future under the new rules," he continued. There can be no question but that the Speedway has had remarkable influence on the development of the high-speed engine and, although regular passenger-car or commercial-vehicle engines do not attain the speed of the racing car engines which have run up to 8000 r.p.m., it is a fact that a large number of our passenger-car engines do reach a speed of 4500 r.p.m. and that this shows the tremendous influence that these racing creations have had on the development of higher speeds. Further, he said, to obtain this great number of revolutions per minute the designers have had to take advantage of the great amount of work which has been done along the lines of using light reciprocating parts and have taught us much about using light pistons, rods and valve mechanisms.

Lessons Learned through Racing

Among those who have, through racing car development, advanced the design of automobiles, Mr. Wall mentioned that racing cars were influential on the advantage of using heat-treated alloy-steels, that most of the tire companies admit that racing has helped very materially in the development of

pneumatic tires, and that the narrow racing cars used during the last few years have demonstrated the great advantage of streamlining any vehicle that cuts the air at a high rate of speed.

Other advantages that have grown from the development of racing cars and racing engines mentioned by Mr. Wall are successful front-wheel drives, the popularization of the eight-in-line engines, the development of the downdraft carbureter and the way in which the difficulties of securing proper engine-lubrication at extremely high engine-speeds. In conclusion, he stated that, regardless of the different opinions formed against the new racing rules, these rules govern the 1930 contest and it is certain on this account as well as others that it will be one of the most interesting races ever run on the Indianapolis Motor Speedway.

The 500-Mile Race a Proving Ground

In the discussion offered by Louis Schwitzer, of the Schwitzer-Cummins Co., he said in part that the race should not exclude ideas of future promise of economical service even though its scope might be confined to motortrucks, motorcoaches and airplanes. Supercharger development in his opinion ultimately would permit the use of smaller engines in large commercial vehicles and yet would supply the same amount of power as larger engines without such equipment. In conclusion he said that, at the next meeting of the Contest Board, specifications will be adopted that will preserve the standard of racing; will not block engineering progress; will allow the driver more leeway with regard to the selection of his car; will induce manufacturers to enter contests will make the race more spectacular from the spectators' viewpoint; and that will make the gate receipts the largest from the promoters' side so that he can award prizes that are worthwhile. Mr. Schwitzer also made remarks expressing his appreciation of having been elected Chairman of the Indiana Section.

Frederick S. Duesenberg, vice-president in charge of engineering of Duesenberg, Inc., Indianapolis, builder of racing automobiles, said that superchargers have caused considerable trouble and that some strife exists between various groups and the racing drivers. On the whole, he feels that it is a mistake to eliminate supercharging or to set a limit of two valves. While the supercharger and several valves per cylinder go hand in hand, he said, nearly as much benefit can be realized by using four valves as when supercharging. The engine speed is not so high but very efficient engines are being built and a well-built stock engine can operate at from 4200 to 4500 r.p.m. and will work with efficiency with each cubic inch of displacement.

Mr. Duesenberg also said that supercharging gives the greatest benefit, not so much in cramming the cylinders, as in breaking up the clean mixture into each cylinder of the entire engine. In his opinion, the best form is a turbo-type supercharger which has a diffuser that runs at speeds of 2000 to 35,000 r.p.m. In conclusion, he remarked on the variety of cars entered for the race and said, with his characteristic humor, "We are going to learn a lot of things, even if we do not learn anything more than that changing the racing rules was a sad mistake."

Tire Development Discussed

Tire development was discussed by E. Waldo Stein, official A. A. A. tire representative from the Firestone factory supplying equipment to practically all cars entered. He told of research which has improved tires so that in the 1929 race they gave minimum trouble and of the adoption of larger sizes for the heavier cars of 1930's competition. The standard size for this year, he said, is 6 x 20 in., whereas, during the last five years, the standard was 5.25 x 20 in.

Pete DePaolo, Phil Shaffer and Lou Moore were the only racing drivers to enter into the discussions, all decrying the return to two-man cars, principally because of the extra hazard to life. DePaolo saw the rules as a change for the better, because the public likes a change, he said.

Proving-Ground Operation

Don Webster Tells the Canadian Section about the G. M. C. Development, Methods and Results

DEVELOPMENT and expansion of the General Motors Proving Ground at Milford, Mich., were traced in an address at the May 21 meeting of the Canadian Section by Don Webster, experimental engineer of the Chevrolet Motor Co., who illustrated his address with lantern slides.

Prior to Mr. Webster's talk, a meeting of the Standardization Committee representing the Canadian industry was held as the outgrowth of the stimulating paper presented by H. D. Allee at the February meeting of the Section and considerable progress was made. Mr. Allee's paper is published in this issue of THE JOURNAL.

Section officers for the coming year were elected as follows:

Chairman—A. S. McArthur, general superintendent of the Toronto Transportation Commission.

Vice-Chairman—Frank Averill, production manager of Durant Motors of Canada.

Treasurer—W. E. Davis, of General Motors of Canada.

Secretary—Warren Hastings, editor of *The Canadian Motorist*.

Mr. McArthur was named to represent the Section at the Summer Meeting at French Lick.

Expensive but Justified

Mr. Webster said that the Proving Ground is a rather expensive development but the results that have been obtained by its operation have justified the cost, although there is no yardstick by which the results can be definitely determined. He traced the development of the Proving Ground from the inception of the idea by Alfred Sloan, Jr., president of the corporation, when a few acres of ground and an old barn used as a garage sufficed, to the present time, when the enterprise comprises 1200 acres and has a large staff of workers.

Last year the Chevrolet Motor Co. operated cars on the proving ground over a total of 1,857,000 miles, according to the speaker, who described in detail the tests to which they were put and presented schedules of adjustments, defects in operation, repairs, and gasoline and oil consumption.

Answers to questions asked in the discussion brought from Mr. Webster the further information that variation in speedometer accuracy ranges from 0 to 6 per cent but that maximum speeds of production cars vary only slightly after the cars have been run in. The 1930 model Chevrolet has a maximum speed, he said, of 62½ m.p.h. Undoubtedly the Proving Ground has influenced the work of the designing engineers, who are constantly wanting to try out new ideas. General laughter greeted his remark that trials on the course served as a guide to what they should not do.

Dayton Hears about Automotive Radio

WHILE the installation of radio sets in automobiles might be considered undesirable on account of the likelihood of its distracting the attention of the driver, L. M. Perkins, of the General Motors Radio Corp., the speaker at the March 12 meeting of the Dayton Section, said this was not the case as the program itself did not distract the driver as much as if he had to carry on a conversation with a

passenger. To prevent accidents caused by the driver looking away from the road the set must be designed so that it has no critical adjustments but can be easily tuned without more than a glance at the dial. In designing the set, which is of course the first problem, some factors, such as sensitivity, selectivity, quality of output and low cost, are similar to the household sets; while special factors, such as rugged design, low battery consumption, automatic volume-control and elimination of spark interference, require attention.

Sensitivity must at least be equal to that of the household set and unless the set can be sold at a low enough figure for the drivers of the cheaper cars to purchase, building an automotive radio set is hardly worthwhile. Preventing shock vibrations is another important feature, as in the automotive sets everything except the tubes, which are flexibly mounted, must be rigidly supported. Automatic volume-control, said the speaker, is essential in the automotive sets.

One of the biggest problems in the automotive radio-set is spark elimination. The primary ignition-circuit cannot be shielded as it guides the signal into the set but requires the use of spark suppressors that tend to stop the oscillations in the secondary circuit. Excessive spark interference is also eliminated in this way. Shielding the secondary circuit is not recommended, said Mr. Perkins, unless absolutely necessary because shielding produces a greater effect on the ignition circuit than the use of resisters. The resisters are a tubular section mounted directly on spark-plugs and also in the ignition circuit between the distributor and the coil. The problem is to get a mechanically rigid resister that either will not break or will not open the circuit if it does.

Mounting of the set and equipment is another problem, but the general arrangement is to mount the set to operate from the front seat, a flexible drive being used. This drive provides a variety of positions for mounting and makes possible locating the set over the engine. The dynamic speaker is expensive and increases the drain on the battery so that the magnetic type is generally used. While most of the cone speakers have been installed under the cowl, the future mounting will probably be in the dome, although this will require a special construction of the automobile and costs considerably more.

Installing the aerial is another rather serious problem and is also expensive. It cannot be put in after the car is delivered because the top deck must be taken off and replaced, which is expensive. If the aerial is placed on top of the car, it must be kept away from the body and yet cover as much space as possible.

Cost of Air Transportation

Rathert Compares Cost of Various Types of Transportation for Oregon Section

DEFINITE figures for the cost of travel by airplane, motor-car, motorcoach and railroad and of transportation by air or road were given in the paper read by G. A. Rathert, general manager and chief engineer of the Breese Aircraft Corp., at the May meeting of the Oregon Section.

This meeting was held May 9 at the Multnomah Hotel in Portland, and followed a dinner and entertainment by an orchestra led by Kenneth Jordan. In the lobby of the hotel was exhibited a Packard roadster, claimed to have a top speed of 115 m.p.h., and a radio was installed so that members of the Section need not miss the latest taxicab news from Amos 'n' Andy.

Speed is said by Mr. Rathert to be the basic reason for the existence of the airplane; manufacturers who use the most improved methods of rapid production cannot be satisfied with ox-cart transportation. Mr. Rathert said that an average charge of about $1.20 per hr. for the time saved by extra-fare railroad trains is a low value for the time that can be saved by an airplane. The value of such time will increase with the earning capacity of the passenger and with the urgency of business negotiations, and in some cases minutes may even mean life or death.

Air travel is more comfortable than railway travel, particularly when consideration is given to the difference in time. The reliability of air travel is approaching that of other modes, but its safety has not been definitely established. A comparison based on the estimated average mileage traveled indicates that injuries are three times as common in automobile accidents and that fatalities are five times as common in airplane travel.

Costs Compiled for Various Airplanes

Cost of airplane travel can only be determined by careful consideration of the items for specific airplanes. Such figures were compiled and presented by Mr. Rathert for five airplanes, ranging from a 60-hp. two-seater with a cruising speed of 80 m.p.h. to airplanes having capacities for 7 and 32 passengers and cruising speeds of 140 and 120 m.p.h. The figures seem to be complete, including, among other items, return on the investment, depreciation, insurance and pilot's salary and bonus. The best available costs were obtained also for automobile, motorcoach, motor-truck and train operation, and three of the airplanes were compared in cost per mile, cost per passenger-mile and cost per ton-mile with the older methods of transportation. He finds the airplane costs reasonable when allowance is made for the time saved.

Analysis of the efficiency of the various airplanes, based on equal speeds, indicated to Mr. Rathert that the highest general efficiency is found in airplanes in the 2500-5000-lb. class and of the strut-braced monoplane type. Large transport planes and biplanes in general he finds to be relatively inefficient.

Mr. Rathert predicts that air transportation will become more and more a necessity and that fares and rates will decrease with improvements and lowering of the cost of the equipment and with increased patronage.

Questions elicited the information that none of the five airplanes studied in the paper was of Breese manufacture. The newest Breese design, a strut-braced monoplane that will be fitted with either a seven-place cabin or a mail fuselage and powered by a 425-hp. engine, has not yet received the type approval of the Department of Commerce. Estimated flying costs for this plane were presented.

Further discussion had to do with features of the Curtiss Tanager plane and with the operation of multi-engined planes with one engine dead. Mr. Rathert said that flight can be maintained with two engines of a three-engine plane, but one of the side engines alone is practically useless. An adjustable fin can be provided which will compensate for the off-center pull enough to enable the pilot to maintain normal control when only one of the side engines is in operation.

Length of Engine Life

Some comparison was made between motor-trucks and airplanes as to proportion of payload to empty weight and as to engine life. H. W. Drake, of the Portland Gas & Coke Co., said that the engine life on trucks far exceeds the 2000 hr. allowed for aeronautic engines, although the truck may be overloaded and required to pull a trailer. Mr. Ogden, of U. P. Stages, Inc., also said that motorcoach engines are fit for more service when the complete vehicles are discarded at the end of four years, after having operated 75,000 to 80,000 miles per year. A. L. Bettys, of Varney Airlines, and others, said that airplane engines usually can be flown at least 3000 or 4000

hr. If the airplane speed is 130 m.p.h., its mileage also is quite high. Airplane engines often are suitable for private or local use after their usefulness for air transport is passed, but it is unsafe to fly with an engine that is so old as to be unreliable.

The Oregon Section is making ambitious plans for a combined technical meeting and outing to be held at Longview Saturday, June 28. Among the papers to be presented is one by R. J. Minshall, of the Boeing Airplane Co. A profusely illustrated program is being prepared, which will be given wide distribution. A regular activity of the Section is a Friday noon luncheon, at which the attendance is usually between 14 and 16. It is planned to hold these luncheons throughout the summer.

Officers were elected for the ensuing year, according to the list presented at a previous meeting of the Section, as reported elsewhere in this issue of THE JOURNAL. H. W. Drake, the new Chairman, is looking for great progress during the coming year as he gets the committee activities under way.

Double Session at Detroit

Welding Society and S. A. E. Section Meet in Afternoon—"Strick" Tells of V-16 Engine in Evening

AGAIN the Detroit Section held a double session at its April 21 meeting, the afternoon session at the Book-Cadillac Hotel being a joint gathering with the American Welding Society to hear an address by H. M. Wofter, of the C. E. Swift Welding Machine Co., on resistance welding, and the evening session following dinner being devoted to a short paper on the Cadillac V-16 engine given by W. R. Strickland, chief engineer of the Cadillac Motor Car Co.

Mr. Wofter first reviewed hastily the accidental discovery of the process of resistance by Prof. Hugh Thompson in 1885 and the retardation of the use of butt and spot-welding by blanket patents covering the process.

Impetus was given to resistance-welding in 1914 by expiration of the blanket patent, the requirement of quick, cheap welding on war materials and the absorption of the infringing company by the parent resistance-welding machinery company. After paying royalties and leases under the Hermata patent for a number of years, a number of automobile companies in Detroit, Pennsylvania and New York State fought the patent and in 1922 or 1923 won a court decision that left the field open. Since then, said Mr. Wofter, resistance welding has become one of the greatest boons ever given to the automotive industry.

Welding Bodies and Axles

As an example of what is being accomplished with this type of welding on a production basis, the speaker mentioned body jigs used by the Edward G. Budd Mfg. Co. in Philadelphia in which the body parts are assembled, the jigs closed, the electric current turned on and the whole body welded at once. He also referred to use of this type of welding in the production of rims at the rate of 24,000 per day in four welding machines. Expert welders are not necessary; the machines can be run by cheap labor.

Millions of rear-axle housings have now been welded in machines that weld the two complete halves in from 4 to 6 sec. Before the development of the Murray welder, almost every other possible welding method had been tried, including gas welding and hand and automatic electric arc-welding. The possibilities of welding each half axle in a two-phase welder are now being investigated. Such a welder, according to Mr. Wofter, can be connected, by means of a Scott transformer, to all three phases of the city current-supply line, thus avoiding the great expense of motor-generator sets for single-phase alternating current for the welder and direct current for the enormous bank considered necessary for use with these welders.

New Projection-Welding Process

Projection welding is a new method that has entered the field within very recent years and which the speaker said is most applicable to the automotive industry, because it applies better to thin flat sheets than to heavier stock. In this process, flat stock is stamped with a few projections so that when two pieces are laid together one rests on the projections of the other. When the pieces are brought between the platens of the welding machine, the projections are welded as in spotwelding, but the surfaces of the pieces are left smooth and without buckles.

This method is so new that not many data have been accumulated, and many formulas are needed. Mr. Wofter suggested that if young welding engineers want to do something for the industry they can do some research work to determine how large an area can be taken care of in a single machine; the greatest number of projections it is possible to make on stock of a given gage; what is the lightest gage sheet that can be projection-welded; what is the heaviest gage; whether the method of punching the projections will strain the metal beyond the elastic limit and whether the weld will have as much strength as a corresponding rivet.

Caution Regarding Cam Shape

Mention was made of the many variables that enter into the matter of shape of the cam used on the welding machine, and the speaker cautioned his hearers against taking too literally some of the statements made and curves presented in published articles on welding. If they try to build a machine to them, they will have a long time in which to regret it, he said.

The address was concluded with the showing of a large number of slides of different types of resistance-welding machines and running comment on their application.

Reasons for the V-16 Engine

The fundamental reasons for the development of the Cadillac 16-cylinder V-type engine and the influence of various requirements on its design were reviewed in the paper presented by Mr. Strickland at the evening session. After reading the paper, a series of charts, photographs and drawings of the engine were shown on the screen.

In addition to the qualities sought with eight-cylinder engines, such as rapid acceleration; high speed; smooth, quiet, easy operation; quick deceleration; riding comfort; character and beauty of appearance; and class and finish of appointments, other factors that affected the engine design were car weight and wind resistance. Both of these have increased in cars in the higher-price class, with a consequent demand for further increase in power.

To meet the conditions adequately, said Mr. Strickland, would require 40 per cent more power. And the question was, How can this be obtained? After canvassing the possibilities, the decision was to go to 16 cylinders. The reasons for this were given by Mr. Strickland and illustrated with charts. Layout of the engine and details of its parts were explained and reasons for the design were given.

In conclusion, the speaker stated that the results include "silkiness" of engine operation, quick pick-up, smooth and quiet running on the road, a compression ratio of 5.5:1, acceleration with seven-passenger sedan body at the rate of 4 ft. per sec. or better, with top speed well over 80 m.p.h., and economy at medium car-speeds equal to that of the eight-cylinder engine.

Carbureters Oregon's April Topic

MANY occurrences in the past have delayed the publication of Section Meeting reports, but the late appearance of the news account of the April 11 meeting of the Oregon Sec-

tion is due to a rather unusual one, the partial destruction of the report by fire in the Air Mail plane that was bringing it to the S.A.E. office. The attendance at the meeting, which was held at the Hotel Multnomah, Portland, Ore., was 55, and a brief business meeting followed the dinner at 6.30 p.m. and preceded the technical session. The principal item at the business meeting was the report of the Section Nominating Committee presenting the names of the following candidates for Section officers for the ensuing year:

H. W. Drake, garage superintendent, Portland Gas & Coke Co., Chairman; C. C. Humber, transportation superintendent, Longview Public Service Co., and Prof. F. G. Baender, head of department of engineering, Oregon State College, Vice-Chairmen; F. P. Myers, president, Myers-Blackwell Co., Secretary, and J. V. Savage, superintendent of the Municipal Shop, City of Portland, Treasurer. With the exception of Mr. Drake, who was a Vice-Chairman for the present Section year, the other officers have been nominated to succeed themselves.

The paper of the evening was presented by G. W. Gleeson, instructor of mechanics and materials at Oregon State College, who collaborated with Prof. S. H. Graf, director of engineering research, in preparing it. The theories of combustion and carburetion are first covered, followed by their application in the design and construction of carbureters. The use of an analysis of exhaust gases to check the adjustment of the carbureter on a motor-vehicle under actual operating conditions was explained, and data showing the savings in fuel consumption effected by the method of adjustment were presented. An interesting discussion followed the presentation of the paper.

Two Subjects at St. Louis April Meeting

AIRCRAFT-ENGINE maintenance and modern piston-ring practice were the two subjects dealt with at the April 15 meeting of the St. Louis Section. Mr. Shedenhelm, of the Parks Airport, told in an extemporaneous address the details of overhauling the Curtiss 0X war-type engine, the recent air-cooled radial engines and in-line engines as practised by the organization with which he is connected. He covered the subject so thoroughly that members in attendance had few questions to ask in the discussion.

A. J. Mummert, chief engineer of the McQuay-Norris Mfg. Co., presented a prepared paper on piston-ring practice, following which the subject was discussed at considerable length by half a dozen of the persons in attendance. The discussion elicited much additional information from the speaker.

Chairman W. L. Dempsey presided.

Solving Differential Expansion

Cleveland Hears of New High-Expansion Aluminum and Low-Expansion Ferrous Alloys

CHAIRMAN W. E. ENGLAND was obliged to be absent from the last meeting of the Cleveland Section during the year in which he has served as its chairman, on May 10, but he sent his regrets, saying that this is only the fourth meeting of the Cleveland Section that he has missed in ten years. After saying that, Mr. England wasted a whole paragraph in saying that he had enjoyed the meetings of the Section and the advantages that he had formed there. In commending the work of the officers during the year past, he mentioned especially that of Benjamin H. Blair for his editing and management of the S.A.E. Junior Journal. In the absence of Mr. England, Vice-Chairman Ferdinand Jehle presided.

Guild Hall, on the tenth floor of the Builders' Exchange Building in Cleveland, was the place of this meeting, which included a dinner and entertainment features; and Frank Jardine, chief engineer of the Aluminum Castings division of the Aluminum Company of America, was the speaker. Speaking on the subject of Thermal Expansion in Automotive Design, he told of a newly developed aluminum alloy which has a coefficient of expansion much lower than that of pure aluminum and its better-known alloys.

While the coefficient of this alloy is higher than that of cast iron, a special nickel alloy of iron has been developed which corresponds so nearly in expansion that a piston and cylinder made of these two alloys need no special provision or extra clearance to permit them to function without piston slap when cold or binding when hot. Lo-Ex is the trade name of the new aluminum alloy, and Ni-Resist of the special iron alloy, which has been developed by the National Nickel Co. The iron alloy contains minimums of nickel, 12½ per cent; copper, 5 per cent and chromium, 1½ per cent, and it is used under some conditions because of its superior resistance to corrosion.

This combination of metals has been under test in an engine running for 200 hr. at 2500 r.p.m. and three-quarters load. The wear on the skirts of the plain trunk pistons was 0.0004-in. and on the cylinder liners was 0.0003-in., and there was no evidence of binding during the test. The liners were placed in an aluminum cylinder-block, used with separate aluminum cylinder-head and crankcase.

Problems of Alloy Connecting-Rods

Thermal expansion has to be considered also in the design of aluminum-alloy connecting-rods. Mr. Jardine showed a diagram indicating the expansion of a big-end bearing of an all-aluminum connecting-rod with aluminum bolts and with steel bolts, and also with a steel cap and steel bolts. This indicated that the vertical expansion was reduced about 20 per cent by means of the steel cap. Steel caps are used on the aluminum-alloy connecting-rods of several automobile engines, to prevent looseness of the bearing when hot. Aluminum-alloy connecting-rods are used in industrial engines also, because the better conductivity of the material prevents the bearing from overheating. Aluminum caps are said by Mr. Jardine to be satisfactory for this class of work. One advantage of the combination of steel and aluminum around this bearing is that the horizontal expansion allows entry of oil at the sides of the bearing without looseness in a vertical direction.

An interesting test was described in which an aluminum and a steel connecting-rod are fitted to a short steel shaft, which is then immersed in oil heated to 500 deg. fahr. It is said that the babbitt will be melted out of the aluminum, but will remain intact in the steel, indicating the superior ability of the aluminum to dissipate heat.

Expansion of Aluminum Cylinder-Heads

Aluminum cylinder-heads present a problem of differential expansion, sometimes being blamed for water leaks at the cylinder-head gasket. No trouble is experienced with the individual heads of sleeve-valve engines, but Mr. Jardine says that it may be advantageous to divide an exceptionally long cylinder-head into two parts. The solution that he commends most highly, however, is to make the whole cylinder-block of the same material as the head. Even the studs holding the cylinder-head in place sometimes cause trouble because of excessive pressure under the washer, which may cause a permanent set and looseness after the engine cools down. The common remedy is to use washers large enough to reduce the unit compressive stress to a safe amount. Other solutions mentioned are the use of aluminum-bronze studs or of differentially expanding spacers.

Aluminum bronze valve seats are recommended for use with aluminum cylinder-heads. Such seats are sometimes cast in place, but Mr. Jardine recommends shrinking them in place as better practice.

Aviation engines are being built extensively with pistons made from the

Lo-Ex alloy, making possible their fitting with approximately 20 per cent less running clearance than when the common aluminum-copper piston-alloy is used. Trunk pistons are said to be a necessity in aircraft engines, and the high-expansion nickel-cast-iron is recommended by Mr. Jardine as a material for airplane-engine cylinders. Liners of this material have been cast into ribbed aluminum cylinders and this combination is recommended for consideration to replace the cast-iron cylinders used in the less expensive aircraft engines.

Production Questions Answered

Several of Mr. Jardine's 70 hearers asked questions regarding methods of shrinking sleeves into aluminum cylinder-blocks, about aluminum connecting-rod design and other subjects. He said that the new alloy he described should be machined with tungsten-carbide tools as a manufacturing proposition, although the use of ordinary cutting tools is practicable. Lyle K. Snell reported some experience with slit-skirt aluminum pistons which demonstrated that they must be carefully machined. An experimental car caused trouble because of sticking rings, and it was found that the ring grooves were not true circles. Another set of aluminum pistons had its ring grooves machined very carefully, so that the dial indicator did not move when placed against the sides of the grooves. The car containing these pistons has been driven 75,000 miles and Mr. Snell avers that a 0.002-in. feeler could not be inserted between any ring and its groove today.

At the close of the meeting, Chairman Jehle announced the result of the election of officers for the Section, for the ensuing year, as follows: D. S. Cole, Chairman; W. E. England, Vice-Chairman; Ferdinand Jehle, Treasurer; and Hoy Stevens, Secretary.

Road-Building in California

State Highway Engineer and Los Angeles Deputy City Engineer Describe Methods

MEMBERS of the Southern California Section were treated to somewhat of an innovation in the matter of subjects presented on the occasion of the monthly dinner of the Section on May 9 at their regular meeting place in the City Club dining-room.

The subject for the evening was The Influence of Motor-Vehicles on Highway Design and Construction. The speakers selected for the evening were T. E. Stanton, material and research engineer for the State Division of Highways of California, and Ralph W. Stewart, deputy city engineer of Los Angeles. Both men are widely known for their work in highway engineering and are regarded as National authorities on matters pertaining to street and road construction.

Although the subject is one that lends itself admirably to technical discussion, Mr. Stanton preferred to give his interested hearers a practical picture of the work which his department and every State highway department has had to do to provide suitable roadways and roadbeds for rapidly changing automotive-vehicle design.

Race Between Engineering Brains

The details of the technical phases of the civil engineers' work in this respect is no doubt frequently overlooked by members of the automotive engineering craft, but a little attention to construction details in any of our highways will show that there has been a considerable race between engineering brains, the brains of the automotive engineer providing greater speeds and loads, and those of the highway engineer providing more substantial subbases for highways, better design and straighter alignment.

One of the most obvious developments in the modernized highway has been that of greater width. Another has been the improved alignment. In support of this, Mr. Stanton cited the development on one of California's most widely known highways — the Ridge Route—saying in part that, "in keeping with the development of the motor-vehicle, we no longer consider a 15-ft. width of pavement safe for our modern high-speed traffic, and seldom construct anything less than a 300-ft. radius curve. On the same route between Bakersfield and Los Angeles, where we originally used a minimum 70-ft. radius, an alternative location is now being constructed with a minimum radius of 1000 ft. Grades are reduced correspondingly to accommodate heavy-truck traffic."

Advance in Construction Practice

The great advance that has been made in construction practice was of special interest to those present. Mr. Stanton told how, with the greater demands of highway use, his department has of necessity introduced methods of testing soils on which roads are to be built and materials of which the surfaces are to be constructed. As outlined, some of the technical work of his department is the investigation of soils from which fills and subgrades are to be made; moisture absorption, swell and shrinkage characteristics, which are studied in the laboratory; borings made in fills where immediate hard-surfacing is proposed and tests to ascertain if full settling or compaction has taken place and tests for shrinkage of clay and adobe soils on which pavement is to be laid. Roadbeds on clay or adobe soil are often opened to traffic for a time after an application of rock or gravel, so that they will become further compacted.

All Materials Carefully Tested

The materials and research department also makes tests of cement to be used in concrete highways and bridges. In the case of concrete highways, tests are also made of gravel and sand, and these materials are tested when they enter into surfacing practice, either in the dry state or when used with oil. Oil for oil-processing and asphaltic soil for black-surfaced pavements are also tested and must conform to certain standards set by the California Division of Highways.

Another duty of Mr. Stanton's department is the testing of metal culvert material and the testing and inspecting of bridge steel. Timber is inspected at the mill and in the field and when necessary is given a special physical test at the laboratory. Paint is sampled and analyzed; expansion-joint materials are tested, and, in short, every physical property that has anything to do with the construction of highways or bridges and culverts is given thorough scrutiny before the material is allowed to enter as an integral part of the highway system. Mr. Stanton said that, during the nine months from July, 1929, to March, 1930, 23,023 tests were made.

Roughometer Shows Greater Smoothness of Present Roads

A highly interesting development of the materials and research department is a device known as the roughometer. Briefly, this measures spring travel of a car as it travels over the road. The value of the device may be noted from the department's figures since 1924, at which time it was first used. Average roughness in inches per mile of cement-concrete pavement was reduced from 19.2 in 1924 to 8.2 in 1929. Asphaltic pavements were reduced from an index of 30.1 in. in 1924 to 13.6 in. in 1929.

The smoother pavements have meant much to the automotive industry and to users of automotive equipment of all classes, appreciably reducing operating costs and adding considerably to riding comfort.

The great interest shown in the general subject, as discussed by Mr. Stanton, was indicated by the large number of questions on various phases following the reading of the paper. Many of

these had to do with the mechanical details of the roughometer and its accuracy and action.

Traffic Demands Outstrip Street Work

The next speaker, Ralph W. Stewart, had previously appeared before members of the Southern California Section at various times. He began his talk with a discussion of the antiquity of the street and highway, and even quoted a passage from the book of Isaiah in proof of the fact that the motor-car was described 3000 years before the first one was produced. His subject was, Road Construction with Special Reference to Problems Relating to City Streets.

Mr. Stewart pointed out that the demands of traffic are still ahead of the ability of engineers to provide roadways and of the various branches of government to pay for them.

Advanced practice in construction was described in considerable detail. The speaker told of recent studies made in the relation of water to concrete mixes and the way in which the city engineering department has adopted new methods of time mixing and other improved practices in the building of pavements.

Following his interesting address, Mr. Stewart was asked several questions on details of street construction and maintenance. At the request of various members, he told of some of the many plans that have been made to improve traffic conditions and also to improve pavement and street design.

Chairman William Fairbanks announced at the beginning of the meeting that there would be a surprise speaker. As a matter of fact, two were introduced. The first was Ethelbert Favary, the Section's perennial and persistent enthusiasist, who gave a 10-min. talk on How To Make Money, Though an Engineer. This encompassed the psychological principles of success.

Highway Improvement in Southern Countries

The other surprise speaker was "Ted" Hopgood, who recently returned to Los Angeles from an extended trip to South America, Central America and Mexico. In line with the topic of the evening, he gave a considerable account of automotive progress and road building in the countries visited. He made a similar trip 10 years ago and therefore could draw some interesting comparisons of that period with the present time. He declared that all of Soutn America is becoming motor-minded and that many of the countries are making remarkable advance in the building of roads.

The movement is particularly noticeable in Mexico, because the money for roads is going where it is intended to go rather than being diverted to other governmental and private ends. This reform was initiated by the Calles régime, which has done more to advance the country than any other influence in recent years.

Mr. Hopgood declared that the people, being of Spanish persuasion, are lovers of ease and comfort and are discovering the greater comfort of improved motor-car in combination with good roads. While development cannot even be compared with that in our own Country, it is evident that the good roads movement has taken root and will probably increase rapidly within the next decade.

New Section Officers Elected

This being the end of the fiscal year for the Section, ballots were countea and announcement made of the officers elected for the ensuing year, as follows: Chairman, F. C. Patton; Vice-Chairman, Wendell E. Mason; Treasurer, J. Jerome Canavan; and Secretary, C. H. Jacobsen. Harold T. Ramsay and Ethelbert Favary were named as additions to the Governing Board of the Section.

The new Chairman has been Chairman of the Papers Committee, and he was thanked by Chairman Fairbanks for his work and given a "hand" by the members. In a short response, Mr. Patton paid high tribute to Mr. Fairbanks and Mr. Favary and spoke enthusiastically of the prospects for the coming year.

The May meeting of the Section was the last regular meeting until October. There will, however, be a summer outing, announcement of which is to be made later. H. B. R.

Ethyl Gasoline Economies

Operating Engine Demonstration and Entertainment Enthuse Northern California Section

AT THE Northern California Section Meeting on May 15, Howard A. Reinhart, sales engineer of the Ethyl Gasoline Corp., discussed the economies resulting from the use of ethyl gasoline with particular reference to the maintenance man's viewpoint. The meeting, which was attended by approximately 150, was held at the Engineers Club in San Francisco, being preceded by a dinner, and before Mr. Reinhart delivered his paper, a short business meeting was held. The principal item at this was the announcement of the election of officers for the ensuing administrative year. Those who will be installed at the August meeting of the Section to guide its destinies for the next 12 months are: Chairman, Dr. Edward Zeitfuchs; Vice-Chairman, Howard Baxter; Vice-Chairman (East Bay) Carl Abell; Secretary, William S. Crowell and Treasurer, Carl Vogt. Mr. Crowell was renominated to succeed himself and Dr. Zeitfuchs and Mr. Baxter were promoted, having served as Vice-Chairman and Vice-Chairman (East Bay) respectively for the current year.

After discussing briefly the various factors influencing detonation and the use of various knock suppressors such as benzol and tetraethyl lead, Mr. Reinhart presented a theoretical comparison of ideal Otto cycles. This was supplemented by data on road and laboratory tests.

In the road test a standard six-cylinder L-head engine was used, a standard head with a compression ratio of 4.7 to 1 and a high-compression head with a ratio of 8.3 to 1 being used. The first test was for hill-climbing ability and was made on a hill 840 ft. long and approximately an 8-per cent grade. With the standard head and ordinary gasoline the car climbed the hill in 33.6 sec. and attained a speed of 23.5 m.p.h. at the top. With the high-compression head and ethyl gasoline the top of the hill was reached in 29 sec. and a speed of slightly more than 30 m.p.h. was attained. In the fuel-consumption test the fuel consumption was slightly less than 20 miles per gal. with the standard head and approximately 23½ miles per gal. for the high-compression head, both results being obtained at a speed of 23 m.p.h. At a speed of 40 m.p.h. the figures were approximately 17 miles per gal. for the standard head and slightly more than 19 miles per gal. for the high-compression head. In another series of tests made on Mt. Diablo where the road makes 209 sharp turns and various grades ranging from 2 to 18 per cent are encountered, the average fuel consumption was 14.1 per cent better with the high-compression head and ethyl gasoline, a saving of more than 7 per cent in the time of climb was effected and gear-shifting was reduced 35 per cent.

Results in Motorcoach Operation

In testing the effect of ethyl gasoline on motorcoach operation two routes were selected. One of these was practically level with no very steep grades, while on the other at least half of the distance was made up of grades varying from very slight ones to others that required considerable running in low gear, the difference in level between the two terminals being 1400 ft. To enable these routes to be covered a demand was made by the operator for more power and high-compression

heads were supplied. After they were installed and several runs made, it was found that not only had the power not been increased, but considerable spark-plug trouble was experienced, the engines heated up considerably and severe detonation was noted. To overcome these difficulties encountered when using ordinary gasoline, the operator decided to make some trial runs with ethyl gasoline. These tests showed an average of 6¾ miles per gal. with the new fuel as compared with 5 miles per gal. with the ordinary gasoline.

While formerly considerable difficulty has been experienced in maintaining the schedule, the change of the fuels eliminated this which, Mr. Reinhart said, could be accounted for by the fact that considerably less gear shifting had to be done and the grades could be negotiated in higher gear than formerly. Boiling in the radiator, which formerly occurred, was entirely eliminated even on the steepest grade and at a time when the atmospheric temperatures were as high as 110 deg. fahr.

Not being thoroughly satisfied with the tests on this 342-mile route, the motorcoach company selected three other vehicles to make similar tests. Two of these covered the low-grade route which had a mileage of 280 for the round trip, while the third vehicle was operated between the same terminals as the route on which the first tests were made but over slightly different roads so that only one trip of 181 miles could be made in a day. In the case of the 280-mile route the mileage per gallon increased from 4⅝ with ordinary gasoline to 5¾ with ethyl gasoline and on the other route an increase from 5 to 6¼ miles per gal. was noted.

Engine Demonstration

Following the presentation of the paper a small Delco-Light generating set was operated at a compression-ratio 6.4 to 1 and a speed of 800 r.p.m., the occurrence of detonation being indicated by the flashing of small lights mounted on the panel board. The engine was first operated on ordinary gasoline until violent detonation occurred, and then without any adjustments, a change was made to ethyl gasoline with the result that detonation was gradually eliminated. Another test was made in which the engine was allowed to run with severe detonation and after shutting off the ignition, the engine continued to run due to autoignition from the high temperature developed. After changing to ethyl gasoline and running the engine until detonation had been eliminated, turning off the ignition resulted in stopping the engine almost at once, proving that combustion temperatures are lower when detonation is eliminated.

The discussion following the demonstration dealt with such subjects as the likelihood of lead poisoning, damage to mufflers, sticky valves and the use of iodine as a knock suppressor.

Premium or Non-Premium Fuels

Dr. Haslam Explains Their Relative Merits at Second Meeting of Baltimore Section

AT THE second meeting of the Baltimore Section, which was held on May 22, the attendance was 76 and a short business session preceded the delivery of a paper by R. T. Haslam, vice-president of the Standard Oil Development Co., explaining the relative merits of premium and non-premium fuels. At the business session the results of the election for officers for the administrative year beginning June 1 were announced. The officers chosen were: Chairman, George O. Pooley; Vice-Chairman, Norton L. Dods; Secretary, Joseph Bavett, and Treasurer, Villor P. Williams, all of whom have been performing the duties of these offices since the recent organization of the Section. Chairman Pooley appointed two standing committees, one on papers and receptions, the chairman of which was A. Bruce Boehm, and the other on membership with Edward W. Jahn as chairman.

Dr. Haslam, the speaker of the evening, began his remarks by defining premium gasoline as a good gasoline with a marked superiority in certain properties and possessing these superlative properties to a sufficient extent to justify an extra price over that paid for what is ordinarily known as a good gasoline. The requirements for a good gasoline were stated as being that it (*a*) should start the engine promptly in the coldest weather, (*b*) should require the minimum possible warming up and the minimum possible use of the choke, (*c*) must permit the engine to develop maximum power at all driving speeds, (*d*) must give good power and acceleration even before the engine is thoroughly warmed up, (*e*) must give knockless engine performance and (*f*) must permit throttling down to minimum speed in traffic without bumping and be free from stalling when idle.

After briefly discussing these points, the speaker said that the refiner's problem was to balance the gasoline so that all of these various points were approximately equal and at the same time make it the best over-all product for the money.

The question of vapor locking was the next point discussed. This was stated as being primarily one of car design, although a gasoline manufacturer must endeavor to produce a gasoline that will operate satisfactorily in all cars irrespective of conditions imposed by the designer. In this connection the results of tests of 21 cars on various gasolines were cited. Of this number 4 would gas-lock on almost any fuel, 6 had a tendency to lock and 11 would not lock under any condition. In this connection, Dr. Haslam cited a case of a car in the $3,000 class, which under usual driving conditions would throttle down to 2 m.p.h., operate very smoothly at that speed and idle continuously with one gasoline, while with the other the minimum speed was 6 or 8 m.p.h., and the car would not idle more than 1½ min. without stalling.

Gum and Sulphur

Gum, said Dr. Haslam, is present to some extent in all gasolines, and in the attempt to get an antiknock fuel some gasolines are so high in gum that after driving 2000 miles the car will begin to lose power due to the clogging of the intake manifold and gum formation on the valve stems causing faulty action of the valves. The gum in a gasoline, he added, should be low enough so that a car can be run for an entire season without any trouble due to blocking up of the intake manifold or intake valves.

Corrosive substances should not be left in the gasoline through improper refining. In general most gasolines keep well within the limit set by the Government, although occasionally some gasolines are placed on the market which are not that good. On this point, Dr. Haslam feels that an owner should be able to assume that when he buys $125 worth of gasoline in the course of a year it will not contain any material which will seriously depreciate the value of a $1,000 or a $2,000 car. Some of the refineries are considering a higher sulphur-content, which he thought probably would give no trouble in 99 out of 100 cars. Crankcase ventilation, which is being built into cars, tends to prevent acid formation in the crankcase and its use has undoubtedly lowered the possibility of sulphur in gasoline damaging cars.

Gasoline should not contain too many heavy ends. Crankcase ventilation has materially helped in that respect so that crankcase dilution is almost unheard of in most cars today. However, if keeping the lubricating oil cool is generally practised, crankcase dilution may again become an important point.

As to whether a driver should buy

premium gasoline or not, Dr. Haslam said that this question was difficult for any one person to answer. The factors that really determine the answer are: Can the driver afford the additional 2 or 3 cents per gal., does the type of car actually require premium gasoline and does the man buying the gasoline consider car life and maintenance as well as merely the cost of the fuel? Buying premium gasoline carries with it a certain assurance that a car is being given the best fuel available. Looking at the premium-fuel question from another angle, the additional cost, assuming a fuel consumption of 600 gal. annually, is $18 and figuring the depreciation on a $1,000 car as $250 annually, the car has a useful life of 4 years. The charge for premium fuel is therefore $72 for the 4 years, and the question for the owner to decide is whether with a premium gasoline he can get the same kind of performance from a $1,000 car as he could from a car $1,072. Dr. Haslam believes that any car manufacturer cannot build into a car for $72 the satisfaction that the average customer can get over a period of 4 years by using a premium fuel.

A lively discussion, which was participated in by a large number of those present, continued until formal adjournment at 11.45 p. m. Even then some were not satisfied and continued talking among themselves for three-quarters of an hour longer.

Problems of Safety in Aviation

Temple Joyce Discusses Its Various Phases at First Meeting of Baltimore Section

THIRTY-THREE members and 13 guests constituted the audience at the April 30 meeting of the Baltimore Section, which was the first regular meeting following the organization of the Section. The speaker of the evening was Temple N. Joyce, vice-president of the Berliner-Joyce Aircraft Corp., who presented a brief outline of the history of aviation, pointed out some of the problems encountered, described their solution and analyzed some of the present-day weaknesses.

Aviation's Three Stages

Aviation development, according to Mr. Joyce, can be divided into three periods, pre-war, war and post-war. In the first stage the technical aspect of the industry was in its infancy and development was participated in in most cases principally by hair-brained individuals who probably did not have the ability to make as great a success in other industries as they could in aviation. Engineering played a very small part in the construction of airplanes in those days, a hatchet and a saw, figuratively speaking at least, being the most important implements.

Military demands at the time of the World War gave considerable stimulus to aviation and developed the airplane into a real military weapon with the machine gun synchronized to fire between the rotating propeller-blades. From this to equipping airplanes for bombing was only a short step accompanied by increased demands for technical development and the training of pilots. The results of the latter was the inadequate training of pilots whose psychology was to take a chance under all conditions and never consider the airplane or their own lives.

Following the war, aviation activities were very much restricted, with the result that a rapid growth in the science of airplane design followed. Group engineering, in which no one man was really responsible for the finished design, was the next step. Under this arrangement an aerodynamic man carried on the research, a stress man passed on the stresses of an airplane under various flying and loading conditions and a weight man and the designers and draftsmen completed the group.

Bankers Evince Interest

The result of Lindbergh's flight was that money was thrown into the industry faster than competent executives were developed. As a result executives, who were perfectly capable in other industries but knew practically nothing about aviation were drafted to head operating companies, or the so-called gypsy pilots were suddenly thrust into positions of authority. At that time the decision as to whether to go ahead or to land under unfavorable weather conditions rested with the individual pilot, who, in nine cases out of ten, had flown in the war and was imbued with the idea of taking a chance.

Commercial aviation, said Mr. Joyce, particularly passenger transport, cannot operate on that basis. Probably the greatest step to stop the foolish flying that had been going on in this Country was the licensing of flying schools by the Department of Commerce on the basis of their curriculum and the thoroughness of their courses. Some of the operating groups have imposed such restrictions on their pilots that reckless flying cannot be indulged in with impunity. Certain fundamental laws of flying have been developed as a result of various aircraft operations during and since the war, said the speaker, and if those laws are observed and if equipment is maintained by a good ground organization, aviation will be considerably safer than at present. However, he does not believe that aviation will ever be as safe as other transportation methods. Every increase in speed in the past has been accompanied by a corresponding increase in risks. While this will be true of aviation, said Mr. Joyce, aviation will make rapid progress in the very near future provided the fundamental laws relating to operation and care are observed, the pilot is under proper control while making a flight, an adequate ground organization is maintained and no attempt is made to have the airplane accomplish more than it can with reasonable safety.

Recent Automotive Developments

FOUR speakers on the latest developments in the automotive industry were heard at the monthly meeting of the Buffalo Section on May 6, which was attended by 75 members and guests. The speakers were G. D. Welding, of the Aluminum Co. of America; V. R. Decrow, of the Decrow Engineering Co.; J. R. Holmes, of the Harrison Radiator Co.; and A. F. Carlson, of the Pierce-Arrow Motor Car Co.

The first part of the meeting was taken up with the election of Section officers for the coming year, who were announced as follows:

Chairman—William Edgar John, of the Buffalo Gasoline Motor Co.

Vice-Chairmen—J. R. Holmes, of the Harrison Radiator Co.; E. H. Oehler, of the Stewart Motor Truck Co.; and W. R. Gordon, of the Fedders Mfg. Co.

Treasurer—A. F. Carlson, of the Pierce-Arrow Motor Car Co.

Secretary—Marsden Ware, of the Curtiss Aeroplane & Motor Co.

Three Vice-Chairmen were elected so that each could look after the work of one of the standing committees—the Membership, the Program and the Reception Committees.

Northwest Has Diesel-Engine Meeting

ROBERT S. TAYLOR gave his swan song as Chairman of the Northwest Section at the meeting held in the Bergonian Hotel, Seattle, on May 9. He reviewed the growth of the Northwest Section since its organization meeting in October, 1929, at which time there were only 40 S.A.E. members in the nearby territory. Of this 40, 26 or 27 were present at the meeting, and now the Section has approximately 100 members. Besides this, the

Northwest Section has helped in organizing the new section in Portland, which now has about 60 members.

The weekend outing in the Cascade Mountains, plans for which were reported in detail in THE JOURNAL for May, has been fixed for June 21 and 22. Carroll C. Humber, of the Longview Public Service Co., presented the invitation of the Oregon Section to attend the two-day meeting to be held in Longview on June 27 and 28. Members of all the Pacific Coast sections are being invited to attend this meeting at which technical papers will be read in addition to sports and sight-seeing trips.

Diesel Engines was the subject of the technical portion of the meeting, and Prof. F. G. Baender, head of the department of mechanical engineering at Oregon State College, was the speaker. The subject had been announced as Diesel Engines and Their Peculiarities, but Professor Baender said that the chief peculiarities of the Diesel engine are that it is without peculiarities such as spark-plugs and electrical wiring. Professor Baender's talk was along the same lines that he uses in his college lectures and was accompanied by lantern slides and a chemical demonstration.

Following Professor Baender's lecture, J. D. Ross, of the Seattle electric lighting company, gave a demonstration of high-frequency alternating current.

At the close of the meeting, the tellers appointed reported the election of officers for the ensuing year according to the slate prepared by the nominating committee as reported in the S.A.E. JOURNAL for May; and Donald F. Gilmore, the newly elected chairman, was called upon to close the meeting.

Winds Up on Gasoline

Milwaukee Section Closes Season with Paper by R. E. Wilson on Fuel Tests

FEATURES of the Milwaukee Section's final meeting of the season were a paper on The Significance of Tests for Motor Fuels, presented by R. E. Wilson, of the Standard Oil Co. of Indiana, and the showing of the Bureau of Mines motion-picture film entitled, The Story of Gasoline. At the request of Chairman A. C. Wollensak, who presided, Mr. Wilson supplemented his prepared paper with a brief discussion of the potentialities of low-grade fuels, a subject in which Milwaukee automotive engineers are greatly interested.

The retiring Chairman, in his opening remarks, gave credit for the success of the Section meetings in the year just ended to the cooperation he had received from the officers and committee chairmen, and expressed his appreciation to Vice-Chairman Frantz, Secretary Debbink, Treasurer Krenzke and Committee Chairmen Eells, Reinhard and Nelsen for their willing assistance.

The election of Section officers for next year resulted as follows:

Chairman—J. R. Frantz.
Vice-Chairman—H. L. Debbink.
Treasurer—Eugene Bouton.
Secretary—Prescott Ritchie.

In announcing Mr. Wilson, Mr. Wollensak stated that he had been a professor at the Massachusetts Institute of Technology and had been associated with the United States Bureau of Mines. He is now assistant to the vice-president in charge of manufacturing of the Standard Oil Co. of Indiana and head of the development and patent department, besides which he is a Past Chairman of the Chicago Section of the Society.

Mr. Wilson, in greeting his audience of about 85 members and guests of the Milwaukee Section, expressed his pleasure at observing the progress the Section had made since his prior appearance before it as a speaker three years ago, and remarked that the meetings are being held in better quarters—in the Milwaukee Athletic Club—and that the attendance was larger and the entertainment better.

What Recent Research Shows

The paper dealt at length with the various tests for determining the properties of motor fuels, particularly in the light of recent extensive research by the United States Bureau of Standards and by various industrial laboratories. He discounted the still prevalent public assumption that the principal difference between different grades of gasoline that is of importance to motor-vehicle users is in mileage per gallon, and that volatility is the most important property of gasoline. Actually, the difference in mileage is a minor matter compared with ease of starting the engine when cold, freedom from knocking and vapor lock, uniform distribution of the mixture, crankcase-oil dilution and corrosion of the fuel-induction system and the crankcase.

Recent research has demonstrated, said Mr. Wilson, that the 10-per cent point on the distillation curve of the American Society for Testing Materials is the best measure of ease of starting and danger of vapor lock, while the 90-per cent point is the best measure of ease of uniform distribution and lack of tendency to dilute the crankcase oil. He commented on work done and in progress on tests for antiknock value and the present lack of a generally accepted standard method of making such a test, but predicted that within another year we shall have a standardized method of determining antiknock value, at least of automobile fuels.

Regarding the formation of gum in gasoline, the speaker, after describing various test methods, referred to recent investigations that indicate that the addition of certain inhibiting agents to the gasoline not only prevents the formation of gum in storage but stops the loss of antiknock properties and very likely will lead to the elimination of the gum problem.

Fuel Gravity not Significant

Specific gravity was once the only test or specification for gasolines and this specification still remains in some State laws, although it has no significance from a practical standpoint, as gravity gives no indication of the distillation range. It is generally true that the customer who selects a high-gravity gasoline gets a poorer product as regards knocking and fuel economy than the man who picks the heavier fuel.

Other tests upon which the speaker commented were the corrosion test; the doctor test, which fails to detect dissolved sulphur, which is one of the most corrosive substances, and color as an indication of gum content, although water-white gasoline may contain large quantities of gum while yellow gasolines may contain practically none.

In conclusion, Mr. Wilson said that intelligent cooperation of the automotive engineer will greatly decrease the need of stringent specifications and prevent the trouble that may result if the specification limits are exceeded. He listed three important ways in which the automotive engineer can improve present conditions. These are designing with more care to prevent overheating of the fuel-supply system; general use of crankcase ventilating means or other measures to prevent condensation of water in the crankcase; and more uniform cooling of the engine and the elimination of local hot-spots.

On its part, the petroleum industry can and will furnish gasoline of higher antiknock value and better antiknock stability without getting into any difficulty from gum content.

Low-Grade-Fuel Prospects

When he had completed the presentation of his prepared paper, Mr. Wilson discussed low-grade fuels and said that the normal price differential be-

tween gasoline and kerosene, under present conditions, is only about 3 cents per gal., and that between gasoline and gas-oil is 5 or 6 cents per gal. Some of this price differential would disappear if rigid specifications covering antiknock quality, gum and sulphur content and odor should be applied to gas-oil or furnace oil, which now differ widely in characteristics. And if the demand for gas-oil should outstrip that for gasoline, the price might well go above that for gasoline.

The principal problems to be overcome in the utilization of the heavier fuels are those of starting, detonation, crankcase-oil dilution and odor. Some of the difficulties are eliminated if the fuels are used in Diesel engines, said Mr. Wilson. These engines may be able to compete in airplanes but it is doubtful if they can do so in motor-trucks and motorcoaches, and the higher cost of Diesel engines is not justified in pleasure motorboats. He predicted that for many years to come the difficulties and inconvenience involved in the use of low-grade fuels will outweigh the doubtful savings in total cost, at least so far as passenger vehicles are concerned.

Following the showing of the Bureau of Mines film depicting the processes of producing gasoline, from the drilling of wells to distribution of the motor fuel, the members plied Mr. Wilson with many questions on points covered in his paper and his discussion of low-grade fuels. A considerable number of the questions and comments related to the low-grade fuels.

Mr. Wilson's paper is scheduled for publication in the July issue of the S.A.E. JOURNAL, and his talk on low-grade fuels, with the discussion, probably will appear in a later summer number.

Dayton Ends Meeting Season

AS ITS concluding meeting of the fiscal year, the Dayton Section held a dinner and inspection trip on the evening of May 14. Twenty-two members attended the dinner at the Engineers Club, after which they adjourned to the plant of the Inland Mfg. Co., where their number was increased to about 60 by other members for an inspection trip through the factory.

Mr. McWhorter, assistant chief engineer of the company, gave a preliminary talk on the method of rubber compounding and the molding of steering-wheels and other automotive parts and on the physical properties of rubber. This address and the trip through the plant were extremely interesting and greatly appreciated by the visitors.

The results of the election of Section officers for the coming year were announced as follows:

Chairman—Carl H. Kindl, of the Delco Products Corp.

Vice-Chairman—F. W. Sampson, of the Inland Mfg. Co.

Secretary—G. W. Frank, of the Materiel Division, Army Air Corps.

Treasurer—A. N. Wilcox, of the Dayton Wire Wheel Co.

The business of the Section was formally turned over to the new officers at a dinner held at the Dayton Engineers Club on the evening of May 23.

Aircraft-Engineering Research Conference

A LARGE group of S.A.E. members, as usual, attended this year's Aircraft-Engineering Research Conference, held at Langley Field on May 13, under the auspices of the National Advisory Committee for Aeronautics and many who attended were profuse in their tributes to those who conducted the conference.

The transportation arrangements included a special boat which left Washington early on the previous evening and reached Old Point Comfort for breakfast at the Chamberlin-Vanderbilt Hotel. No details that make for an interesting and enjoyable trip were forgotten. Even a kindly fate seemed to join forces with J. F. Victory, Secretary of the N.A.C.A., by directing a group of vivacious college girls on to the same boat with this party of hitherto austere engineers.

Upon arrival at Langley Field an open technical session was held at the Officers' Club. Joseph S. Ames, Chairman of the Executive Committee of the N.A.C.A. presided at the meeting and Military Commandant Weust, of Langley Field gave the address of welcome.

Reports and graphic presentation of results of work carried on at the various divisions of the Research Laboratory, during the last year were given. Henry J. E. Reid is in charge of the Research Laboratory which has departments covering Flight Research, Atmospheric Wind Tunnel, Variable Density Wind Tunnel, Power-plants Laboratory.

Following luncheon at the Officers' Club inspection tours were conducted including visits to the airfoil machine that cuts the metal from a hand-made wooden pattern and is said to have an accuracy of within one-thousandth of an inch; visits to the Atmospheric Wind Tunnel; the Variable Density Wind Tunnel; the Power-plants Laboratory, where work on spray research and Diesel engines is in progress; the Flight Research Laboratory where the experiments with cowling are going forward; the Propeller Research Laboratory, and the large wind tunnel.

The Conference was called at 2.30 at which time comments, questions and suggestions for future work were in order. Later in the day an opportunity was afforded to visit the excavation and work that is going forward on the new model basin and a race was staged between two Navy Corsair planes, one a standard airplane, the other equipped with N.A.C.A. cowling. The cowled plane was notably the faster.

The total attendance at the Conference was 172, of whom 66, or nearly 40 per cent, are members of the S.A.E.

Personal Notes of the Members

Prof. R. M. Anderson Retires

Announcement has been received that Robert Marshall Anderson, for more than 20 years professor in the department of mechanical engineering at Stevens Institute of Technology, in Hoboken, N. J., will be retired at the end of this college year.

Born in 1862, Professor Anderson was graduated from Stevens Institute in 1887, with the degree of Mechanical Engineer. He was assistant to the factory superintendent of the Spring Torsion Balance Co., of Jersey City, N. J., until the following year, when he took up test and experimental work at Stevens Institute, working in the departments of mathematics and of engineering physics and acquiring considerable experience in the field of internal-combustion engines.

From 1898 to 1901, Professor Anderson was a member of Anderson & Murphy, of New York City, consulting and contracting engineers, and during the next three years held the position of vice-president and constructing engineer of the Bacon Air Lift Co., also of New York City.

For the following four years he was treasurer and consulting engineer of the Hudson Engineering Co., of New York City, cooperating in the development of a double-acting two-cycle engine for automotive service. Following this connection, he accepted the call to rejoin Stevens Institute and remained associated with the faculty ever since.

Professor Anderson has been a Member of the Society since 1914. He was a member of the Research Division of the Standards Committee in 1913 and 1914, and representative of the Society on the Sectional Committee on Scientific Engineering Symbols and Abbreviations in 1926, of the Subcommittee on Aeronautic Symbols and of Subcommittee No. 6 on Mathematical Symbols in 1926.

Plimpton with the White Co.

R. E. Plimpton has terminated his connection with *Bus Transportation*, of which publication he had been associate editor since its first issue in 1922, to undertake certain transportation engineering duties in the organization of the White Co. of Cleveland, particularly in connection with motorcoach work.

Mr. Plimpton's early work included service as detail draftsman with the Fenn Machine Co., of Hartford, Conn., in the summer of 1906, and for the next two years in the operating department of the New York Edison Co. He was graduated from the Brooklyn Polytechnical Institute in 1910 with the degree of Electrical Engineer, and subsequently took the student course in engineering as an employe of the General Electric Co., at West Lynn, Mass. From June, 1912, to March, 1915, he was connected with the General Vehicle Co., of Long Island City, N. Y., being engaged in technical work along service, educational and kindred lines.

In 1915 Mr. Plimpton became affiliated with the publications of the Society, and two years later was appointed field secretary of the Society. These activities were interrupted by the World War, when, in 1918, he took charge of the statistical branch of the engineering division of the Motor Transport Corps., U. S. A., in the City of Washington. Subsequently he worked with the George H. Gibson Co., and with the Newell-Emmet Co., in New York City, and in 1921 entered the employment of *Bus Transportation*.

Mr. Plimpton became a Member of the Society in 1916, and joined the Metropolitan and Chicago Sections in 1921 and 1928 respectively, serving as Secretary and Chairman of the former during 1923 and 1927 respectively. He was a member of the Sections Committee of the Society in 1926 and 1928, and Chairman of the Operation and Maintenance Committee in 1927 and 1928.

Papers by Mr. Plimpton were presented at meetings of the Society and published as follows: Some Fundamental Characteristics of Present-Day Buses, THE JOURNAL, August, 1922, p. 163, and TRANSACTIONS, part 2, p. 532; An Analysis of Costs for Ten Years of Fleet Operation, THE JOURNAL, May, 1924, p. 539, and TRANSACTIONS, part 2, p. 665; Field for the Motor Transport Engineer, THE JOURNAL, May, 1927, p. 595; the Large-Scale Operator's Influence on Design and Construction, S.A.E. JOURNAL, January, 1928, p. 65, and TRANSACTIONS, part 1, p. 257; Operation Activities Outlined, S.A.E. JOURNAL, December, 1928, p. 623; and Long-Distance Passenger Services, S.A.E. JOURNAL, September, 1929, p. 285.

Lon Smith Joins Hercules

After having been established as sales counsel in Indianapolis for several years, Lon R. Smith recently became affiliated with the Hercules Motors Corp., of Canton, Ohio. As assistant director of sales for this firm, he handles export and industrial sales and also is responsible for the direction of sales promotion.

Mr. Smith has been a Member of the Society since 1911. He was a Councilor during the years 1922 and 1923. He served as Chairman of the Indiana Section in 1914, 1917 and 1921, and as Treasurer of the Mid-West Section in 1918.

At a recent meeting of the Aeronautical Chamber of Commerce of America, Inc., the following S. A. E. members were elected to serve on the Board of Governors: **E. L. Cord, H. M. Hanshue, Paul Henderson, Charles L. Lawrance, Edward S. Evans, L. V. Kerber, J. C. Hunsaker, E. E. Aldrin, Charles C. Colvin, E. E. Wilson, J. R. Fitzpatrick, John R. Cautley, Carl B. Fritsche, W. E. Metzger,** and **William P. MacCracken.** Mr. Cautley was elected secretary, B. F. Castle was elected treasurer, and Luther K. Bell was re-elected general manager. Mr. Hanshue was elected vice-president of the southwest section; Mr. Henderson of the north central; Mr. Evans of the east central, and L. V. Kerber of the south central section.

Dr. Miller McClintock, director of the Albert Russell Erskine Bureau for Street Traffic Research, Harvard University, will present the subject of traffic regulation, adaptation of roads to traffic in built-up areas, and parking and garaging of vehicles, as one of the seven general reporters who will present the questions on the agenda of the Sixth International Road Congress to be held in Washington next October. **Roy D. Chapin,** of the National Automobile Chamber of Commerce, is president of the American Organizing Commission. It is expected that delegates from more than 50 nations will be in attendance at the congress.

Coincident with the merger of the Baldwin Chain & Mfg. Co., of Worcester, Mass., and the Duckworth Chain & Mfg. Co., of Springfield, Mass., it was announced that **Frank J. Wechsler,** of Worcester, was to be vice-president and treasurer of the new company, to be known as the Baldwin-Duckworth Chain Corp. **W. F. Cole,** also of Worcester, was also to be a vice-president of the new company.

Walter D. Appel, former chief engineer with Vauxhall Motors, Ltd., of Luton, Bedfordshire, England, is now product engineer with the General Motors Export Co., of New York City.

(*Continued on p.* 44)

Applicants Qualified

The following applicants have qualified for admission to the Society between April 10 and May 10, 1930. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate; (S M) Service Member; (F M) Foreign Member.

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Notes and Reviews

These items, which are prepared by the Research Department, give brief descriptions of technical books and articles on automotive subjects. As a general rule, no attempt is made to give an exhaustive review, the purpose being to indicate what of special interest to the automotive industry has been published.

The letters and numbers in brackets following the titles classify the articles into the following divisions and subdivisions: *Divisions*—A, Aircraft; B, Body; C, Chassis Parts; D, Education; E, Engines; F, Highways; G, Material; H, Miscellaneous; I, Motorboat; J, Motorcoach; K, Motor-Truck; L, Passenger Car; M, Tractor. *Subdivisions*—1, Design and Research; 2, Maintenance and Service; 3, Miscellaneous; 4, Operation; 5, Production; 6, Sales.

AIRCRAFT

Equipment Used in Experiments to Solve the Problem of Fog Flying. Published by the Daniel Guggenheim Fund for the Promotion of Aeronautics, Inc., New York City; March, 1930; 57 pp., illustrated. [A-1]

A report of the Fund's activities in solving the problem of fog flying was issued in October, 1929, and reviewed in these columns in the December issue of the S.A.E. JOURNAL. The present pamphlet has been issued as a supplement to the earlier report to make available the details of methods and instruments utilized in the fog-flying experiments. The operation and essentials of equipment used in the tests are described and pictured, and suggestions made for future developments.

The Design of Plywood Webs for Airplane Wing Beams. By George W. Trayer. Report No. 344. Published by the National Advisory Committee for Aeronautics, City of Washington, April, 1930; 17 pp. [A-1]

This report of the Forest Products Laboratory deals with the design of plywood webs for wooden box-beams to obtain maximum strength per unit weight. A method of arriving at the most efficient and economical web thickness, and hence the most suitable unit shear stress, is presented, and working stresses in shear for various types of web and species of plywood are given. The questions of diaphragm spacing and required glue area between the webs and the flange are also discussed.

Pressure Distribution on the Tail Surfaces of a PW-9 Pursuit Airplane in Flight. By Richard V. Rhode. Technical Note No. 337. Published by the National Advisory Committee for Aeronautics, City of Washington, April, 1930; 13 pp., illustrated. [A-1]

This note presents the pressure-distribution data obtained on the tail surfaces of a PW-9 airplane in a number of flight maneuvers. The results given are a part of those obtained in an extensive investigation of the pressure distribution over all of the lifting and control surfaces of this airplane. They are given in tabular and curve form, and are discussed briefly in respect to their comparison with the existing tail-surface design specifications.

This information has been issued in advance of the report on the complete investigation because of the immediate interest in tail-surface loads occasioned by the results of recent tail-surface pressure-distribution tests and failures occurring in flight. It should serve as a guide to those designers who desire to ensure structural safety in their airplanes pending formulation of more satisfactory tail-surface design rules.

Experimental Investigation of Aircraft Propellers Exposed to Oblique Air Currents. By O. Flachsbart and G. Kröber. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, Dec. 14, 1929. Technical Memorandum No. 562; 18 pp., 30 figures. [A-1]

Balanced and Servo Control Surfaces. Reprinted from *The Aeroplane*, Feb. 26, 1930. Technical Memorandum No. 563; 15 pp., illustrated. [A-1]

Recent Tests of Tailless Airplanes. By Alexander Lippisch. Translated from *L'Aérophile*, Feb. 1-15, 1930. Technical Memorandum No. 564; 10 pp., 9 figures. [A-1]

The above listed three Technical Memoranda were issued during April by the National Advisory Committee for Aeronautics, City of Washington.

Wind-Tunnel Tests of Venturi-Type Cowls and Engine Nacelles Suitable for Multi-Engine Airplanes. Air Corps Information Circular No. 647; 15 pp., illustrated. [A-1]

The Calculation of the Natural Frequency of a Cantilever Monoplane Wing. Air Corps Information Circular No. 649; 11 pp., illustrated. [A-1]

Instructions for Assembly of Detachable-Blade Propellers. Air Corps Information Circular No. 648; 5 pp., illustrated. [A-2]

The three foregoing Information Circulars were issued on March 1, 1930, by the Chief of the Air Corps, City of Washington.

Stossversuche on Druckgummifederungen für Flugzeugfahrgestelle. By K. Hohenemser. Published in *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, March 28, 1930, p. 133. [A-1]

The pressure type of rubber shock-absorber, because of the action of hysteresis, is said to be more suitable for use on aircraft landing-gear than the tension type. Reference is made to descriptions previously published of English pressure shock-absorbers and static pressure tests made on them. The question is raised whether such tests furnish a suitable basis for judging the performance of these shock-absorbers, and this article presents as a further contribution to their study the results of dynamic impact tests.

An analysis of the action of pressure-type shock-absorbers is presented and suggestions involving the simplification of their structure are made. The research described seeks the answer to two questions:

(1) How does the elastic action of the rubber and its hysteresis vary in their manifestation during periods of dynamic deformation such as is experienced during the impacts of aircraft landing-gear from that shown in static tests?

(2) How does the external frictional force vary?

The test apparatus is described and the results of tests on shock-absorbers of different heights and inside and outside diameters are given. Inquiry is made into the type of impact to be expected in practice, and the correct design and suitable choice of an absorber are discussed.

Motorschonung durch Drosseln. By Schatzki. Published in *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, April 14, 1930, p. 164. [A-1]

The potentialities for interruption in the continuous smooth performance of an aircraft engine are many: failure of material, poor repair or maintenance service, and unsuitable fuel or oil, to mention a few. These potentialities become realities chiefly through the unremitting strain on the engine to give its last ounce of power. On the basis of this hypothesis, the author suggests as a remedy throttling the engine to relieve three sources of stress: high speed, torque and power. He proposes to investigate the effect of relative throttle openings on the average life of

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Notes and Reviews

Continued

an engine, its liability to failure and its liability to cause a forced landing.

The difficulties and expense of such an investigation carried out experimentally are set forth and a statistical study of the operating figures of the Deutsche Luft Hansa is offered as a substitute. Fifteen engines, divided between two types, are covered by the figures. They are divided into three service groups, according to the severity of the conditions they were called on to face. Operating results from Jan. 1, 1927, to Sept. 30, 1928, are presented and analyzed.

While such a statistical study cannot lay claim to the accuracy of an experimental research, it is valuable in confirming the assumption that throttling an engine has a decided influence on its durability and that decreasing the stress on engines will entail decided advantages.

Der Gegenwärtige Stand der Heliumgewinnung und Heliumforschung. By Helmut Beelitz. Published in *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, March 14, 1930, p. 109. **[A-1]**

Hydrogen has not yet, and will not for some years to come, surrender its place as an airship gas wholly to helium. Not only is hydrogen procurable in all parts of the world, but it has better lifting capacity and is cheaper than helium, all of which conditions are recommendations to commercial users. Furthermore, full confidence can be placed in the safety of a hydrogen-inflated ship that is properly constructed. However, the time will come when all aircraft will be obliged to use a non-combustible gas, for the public, educated to the advantages of helium, will come to class as second-rate ships using other types of gas.

These are the conclusions drawn by the author from this study of helium production and research, in which he touches on the peological, industrial, physical and chemical aspects of the question. He reviews the facts known about the characteristics of helium and the history of its discovery.

In surveying the sources of helium, the author points out that during the last year many places have been discovered all over the world where the gas is present mingled with other gases. No geological justification supports the assumption that North America has a monopoly of the valuable material; the fact that commercially usable quantities have been produced only in the United States and Canada is due only to lack of interest and capital available for development in other countries. From two German sources the gas mixture is as rich in helium content as in the well-known American productive centers, but the absolute quantity is small. Still this may be taken as an indication that some considerable European source may be found. The possibilities of helium production from radioactive minerals are also reviewed.

Included in the article are a review of helium production in the United States, practical considerations affecting its use as compared with that of hydrogen, developments in France and Canada, and recent physical and chemical researches in Germany.

Progress. A Pictorial Review in *Flight* Photographs. Published in *Flight*, Jan. 3, 1930, p. 31. **[A-3]**

This review of 21 years of aircraft progress consists largely of a selection of *Flight* photographs of pilots, events, aircraft engines, airplanes, seaplanes, airships and air transport. The collection is remarkably complete and well arranged and should serve not only as interesting glimpse of the past but as a handy reference.

Luftpolitische Jahresschau 1929. By Wulf Bley. Published in *Deutsche Luftfahrt*, January-February, 1930, p. 3. **[A-3]**

What influence has a nation's internal and foreign policies

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COMPRESSION
QUALITY
SINGLE SLOT
DRAINOIL
DOUBLE SLOT
DRAINOIL

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on the development of its aviation? What have been the trends in these particulars during 1929, especially as regards Germany? These two questions are discussed elaborately, from the German viewpoint, in the present article.

So far as its utility for national defense is concerned, the author deplores, the situation of German aviation is as sad as it is easy to understand, and he refers to the terms of the Versailles treaty in explanation of this statement. The economic and political thought of the nation has, since the war, been in a state of ferment, its reasoning based not so much on real and actual necessities as on impractical ideology. Aviation has been dragged into the maelstrom of partisan politics; one association that has been formed has for its professed object the democratization of aircraft; that is, the use of aviation as an instrument for world peace through the bringing together of the proletariat of the various nations. The author decries the idea that aviation, or any means of transportation, works toward general pacification; rather, through facilitating the movement of men, munitions and supplies, it makes war more bitter and widespread.

Turning toward other nations, the author refers to the large sums of money being spent on aviation, and instances expenditures in France, Great Britain, Italy and the United States. He discusses the effect that colonial possessions or colonial ambitions have had on the development of air travel, particularly as regards the exploitation of Africa and the Far East, and the reactions that such trends have had on international understandings. He distinguishes three groups in world aviation; Great Britain and her colonies, the Americas, and middle and western Europe. Germany, belonging to the last group, should attempt to strengthen her position, particularly through the establishment of long air mail and freight routes, which presupposes the development of night and instrument flying.

Le Cinéclinographe Estienne-du Cluzel. Published in *L'Aéronautique*, March, 1930, p. 89. **[A-4]**

The Cinéclinographe is an ingenious instrument, or rather package of instruments, for the measurement of aircraft flight characteristics. It is intended to render ground measurements of speed unnecessary, to eliminate from readings the inaccuracies due to air movements, to provide a means for protecting instruments from disturbance or injury and to provide at one time and in one place all data for the judgment of aircraft performance.

Specifically, the device registers photographically on the one diagram in rectangular ordinates simultaneous values of flying speed and climb, air pressure and temperature, engine speed and time. Its exterior is a tapered, streamlined case provided with tail surfaces that control the axis of the instrument with relation to the wind and dampen oscillations which occur accidentally. The separate instruments lodged in this case are said to be of special interest, not only because of their high precision, but because of their small size and the way in which they are fitted into the space available. The transmission of readings from the instruments to the recording sheet is accomplished through an arrangement of mirrors and light rays. The curves on the diagram can be distinguished from one another by the thickness of the line used and by their general character. A full description of all details of the device is given.

CHASSIS PARTS

The Coefficients of Friction between Rubber and Other Materials. By R. Ariano. Published in *Rubber Chemistry and Technology*, January, 1930, p. 67. **[C-1]**

In the tests that provide the basis of this report, static friction between rubber tires and other surfaces was measured by placing the tire on a wheel held against rotation and fixed to a beam pivoted at one end. The tire rested on

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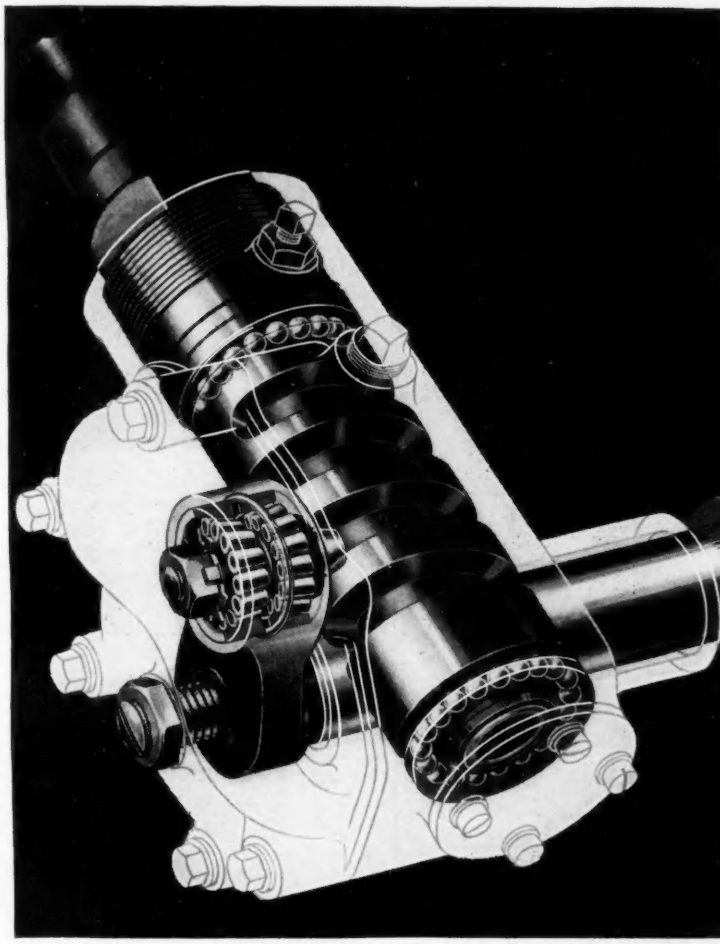

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a horizontal iron plate free to move horizontally but not vertically, and was pressed down on this plate with a predetermined force by loading the beam. A horizontal force was then applied to the plate and increased gradually until it reached a value, F, just sufficient to overcome the friction between the tire and the plate, and thus caused the latter to move. Tests were made with the force, F, acting in the plane of the wheel, that is, the direction of travel, and perpendicular to the plane, corresponding to the forces involved in lateral skidding. In some experiments this plate was replaced by a slab of an actual road-surfacing material.

The results are given in a series of tables from which the following conclusions are drawn:

(1) The coefficients of friction of rubber tires on dry non-dusty surfaces are virtually independent of the load on the wheel and, with pneumatics, of the inflation pressure; on muddy surfaces the coefficients tend to decrease with increasing load.

(2) Dust, mud or water reduces the friction with rubber tires but not with iron tires.

(3) The tread pattern reduces the friction on dry surfaces but increases it on muddy surfaces.

(4) There is no systematic difference between pneumatic, cushion and solid tires as regards coefficient of friction; the details of individual design and material are the deciding factors.

(5) There is no simple relationship between the coefficient of friction and the compressibility or area of contact of the tire.

(6) The coefficient of friction depends on the type of road surface, its deformability, and especially on the presence or absence of dust, mud or water.

(7) Rubber tires have a much higher coefficient of friction than iron tires, especially on dry, hard surfaces.

(8) The static friction is 10 to 20 per cent higher than the dynamic friction.

Nutations et Résonance Gyrostatiques. By M. Broulhiet. Extract from the Mémoires de la Société des Ingénieurs de France, Bulletin July-August, 1929; 109 pp. [C-1]

Gyrostatic action is here analyzed and the analysis applied to the manifestation of and remedies for automotive front-wheel shimmy, faulty automotive suspension and the rolling of ships. All of these three problems of applied mechanics, states the author, can be approached through the theory of gyrostatic movement; and all can be solved through the same equations, characterized by the same five constants, which, however, have different values in each case.

Emphasizing the importance of shimmy, the author states that in the disturbance caused by it considerable energy is manifested and that the force necessary to maintain it amounts to several horsepower. He concludes from his analysis that, to overcome shimmy at all speeds, steering must be fixed, tire construction improved and lateral forces of the vehicle suppressed. To remove the critical speed of shimmy outside the speed range of the car, he recommends increasing the moment of inertia of the entire front axle by adding to it either in whole or in part that of the vehicle. This can be brought about by the system which, he says, is wrongfully known as independent wheel suspension and which he terms "wheels parallel to the plane of symmetry of the vehicle."

In suspension and in steering, the author says, in stating the theme for the second part of his discussion, the real problem is one of horizontal movement. Two vertical movements also involved are the alternative vertical displacement of the center of gravity of the vehicle and its angular perpendicular displacement around the center of gravity.

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His reasoning leads him to the conclusion that a gyrostat is the best shock-absorber, since it, unlike any other absorber, does not detract from the flexibility of the suspension. The author has developed an experimental device of this type for automotive use.

ENGINES

Fuel-Research Set with Variable-Compression Engine. Published in *Engineering*, Jan. 10, 1930, p. 44. [E-1]

This article describes and pictures a knock-testing engine and equipment developed from a standard B-type Armstrong engine in collaboration with the research department of the Anglo-Persian Oil Co. in England. It is said to embody the results of several years' experience with fuel-testing plants and long, intensive work in the development period.

The engine is of the variable-compression type and, while it is pictured with the bouncing pin, it is designed to employ any of the published methods of measuring the anti-knock rating of fuels.

The carbureter has no float chamber, as the use of a float was found to be likely to cause erratic running and, in addition, increased the waste fuel space. The air inlet to the carbureter is fitted with a heating element. The water-circulating system is divided so that the water in the head and in cylinder jackets can be maintained at different temperatures.

The author states that these sets are being adopted by a number of large oil companies in England and other countries.

Roller Bearings. By T. W. Cooper. Paper presented before the Institution of Automobile Engineers, London, England, January, 1930. [E-1]

The perfection of the roller bearing is attributed by the author to the development of the motor-car to its present position of speed and reliability, together with the growth in size of the motorcoach and the consequent increased bearing loads. The various types of roller bearing are described and illustrated, the antifriction bearing is discussed and the relative efficiency of the ball and roller bearing considered. Test results giving the coefficient of friction for ball and roller bearings at various loads and speeds are given in tables accompanying the article. The results of crushing tests on solid and hollow rollers are shown graphically, and compression and creep are considered.

Combustion in Diesel Engines. By H. R. Ricardo. Paper presented before the Institution of Automobile Engineers, London, March, 1930. [E-1]

The author, who is well known for his engine and fuel research work, outlines the trend in high-speed Diesel-engine design and analyzes the combustion process, comparing the Diesel and the gasoline engine through the various stages of ignition, carburetion and combustion. The results of tests made at the author's laboratory over a period of 8 years on a number of sleeve-valve engines are reported and presented diagrammatically.

The Design and Development of an Automatic Injection Valve with an Annular Orifice of Varying Area. By William F. Joachim, Chester W. Hicks and Hampton H. Foster. Report No. 341. Published by the National Advisory Committee for Aeronautics, City of Washington, 1930; 10 pp., illustrated. [E-1]

The injection valve described in this report was designed and developed at the Langley Memorial Aeronautical Laboratory of the National Advisory Committee for Aeronautics in connection with a general research on aircraft oil-engines. The purpose of this investigation was to provide an automatic injection valve of simple construction that

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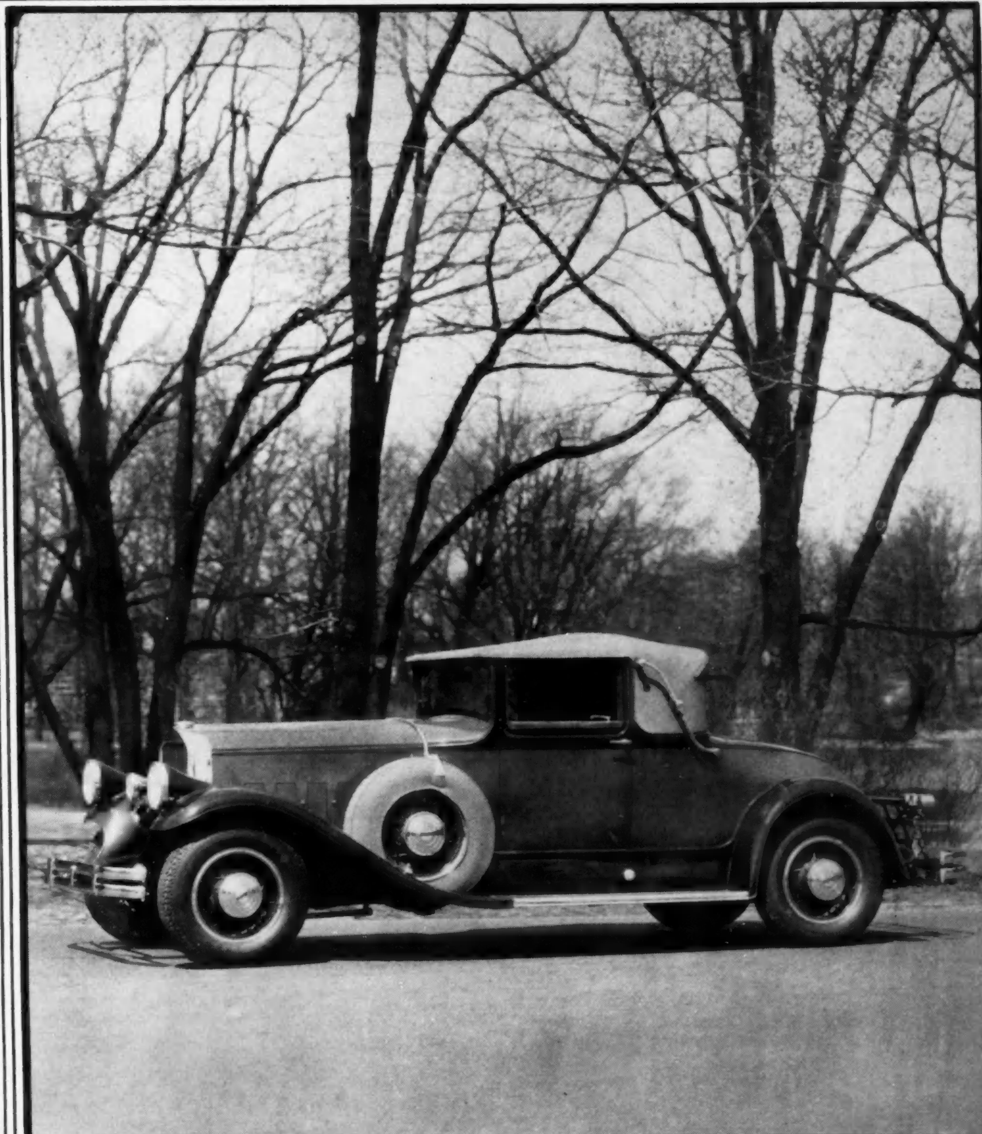

There are 29 years between the two Pierce-Arrows pictured here. Each bears the distinction of being the finest motor car of its day.

THE 1930 PIERCE-ARROW

PIERCE-ARROW, whose span of years closely parallels that of the S. A. E., extends cordial greetings to this distinguished society on the occasion of its Silver Jubilee.

PIERCE-ARROW

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would produce a finely atomized oil spray of broad cone angle and fulfill the requirements of fuel injection in aircraft oil-engines.

The injection valve designed has only six parts; namely, two concentric nozzle-tubes flared at one end, two body parts and two nuts. The nozzle tubes are provided with seats at the flared ends to form an annular orifice which automatically varies in area with the injection pressure. Adjustment of the nuts determines the valve-opening pressure. The fuel passage to the orifice is provided by the clearance space between the nozzle tubes. When sufficient oil pressure is developed by the fuel-pump, the flared ends of the nozzle tubes move apart slightly and the oil passes through the annular orifice, producing a broad conical spray. The nozzle tubes are so constructed as to cause the cylinder gases to heat them to approximately 500 deg. fahr., which preheats the oil and tends to reduce the ignition lag.

The results of tests of this injection valve with the N.A.C.A. spray-photography equipment indicate the effect of several factors on spray penetration. Curves are presented showing these effects, together with the effect of engine-operating temperature on the valve-opening pressure.

Analyses and engine tests indicate that the fuel spray from this type of injection valve has characteristics which reduce the time lag of autoignition and promote efficient combustion in high-speed oil engines.

On the Resistance Experienced by a Cylinder Moving in a Channel of Finite Breadth. By Susumu Tomotika, Rigakusi. Report No. 58. Published by the Aeronautical Research Institute, Tokyo Imperial University, Japan, March 1930; 38 pp., diagrams. [E-1]

In the introduction the author outlines the general trend of recent literature on this subject. He states:

> In a recent paper H. Villat has put forward the theory of the resistance experienced by a cylinder moving in a channel of finite breadth. To obtain the general expression for the resistance, Villat adopted the well-known method of calculation which The. v. Kármán used long ago in his calculation of the resistance of an infinitely long cylinder moving in an unlimited ideal fluid.
>
> It is curious, however, that Villat's general expression for the resistance does not give the well-known Kármán's formula in the limit when the breadth of the channel becomes finite.
>
> On the other hand, L. Rosenhead has also discussed, in a quite recent paper, the same problem independently and has attained the expression for the resistance by making use of Synge's method of calculation. It is shown, in a few lines, that his general expression degenerates into the well-known Kármán's in the limiting case when the breadth of the channel becomes infinitely large.

In his study of the problem the author has applied both Kármán's and Synge's methods of calculation, and has been able to prove that the general expression for the resistance calculated by Kármán's method is the same as that obtained by Synge's method.

Further, since it seemed interesting to see whether the general expression for the resistance would degenerate into the well-known Kármán's resistance formula in the limit when the breadth of the channel becomes infinite, the limiting form of the expression was fully computed and it was found that the general formula gives correctly that of Kármán's in the limiting case.

The Aircraft Engine Instructor. By A. L. Dyke. Published by the Goodheart-Willcox Co., Inc., Chicago, 1929; 425 pp., illustrated. Price $5. [E-3]

As in the previous edition of this volume, also reviewed in these columns, the object of the author is to familiarize the student, mechanic and others mechanically inclined with the principles of construction and operation as well as the maintenance of aircraft engines and their accessories.

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Notes and Reviews

Continued

To accomplish this purpose, the author has confined the greater part of the book to the design, construction, operation and maintenance of such engines as the Wright Whirlwind, Pratt & Whitney, Wasp, Curtiss, Packard and others. In other chapters the general characteristics of carbureters, magnetos, starters and generators, and instruments and controls are discussed.

General aeronautical information is contained in the last two chapters. The first of these offers the names of aeronautic publications and aeronautic clubs; the requirements in aircraft operation; the procedure to follow in applying for airplane, pilot's or mechanic's licenses; and other data. The last chapter presents a dictionary of aeronautical terms and symbols, reprinted from a report of the National Advisory Committee for Aeronautics.

Some of the topics on which data are given in an addendum, lacking in the previous volume, are: installation of modern airplane powerplants; propellers; engine troubleshooting charts; superchargers and rotary induction; fuel oil versus gasoline; and specifications of commercial airplanes, seaplanes and aircraft engines.

Proceedings of the Third Oil Power Conference held at the Pennsylvania State College, June 24 to 27, 1929. Technical Bulletin No. 8. Published by the Pennsylvania State College, Pa., November, 1929; 202 pp., illustrated. [E-3]

The following papers of interest in the automotive field are collected in this issue of the Proceedings:

High-Speed Diesel-Engine Design. By Otto Nonnenbruch, I. P. Morris and De La Vergne

High-Speed Oil-Engine Pumps and Injection Valves. By John L. Goldthwaite

Combustion in High-Speed Oil Engines. By W. F. Joachim

Commercial Applications of High-Speed Oil Engines. By C. H. Gibbons

Diesel Education. By Roswell H. Ward

Some Results of the Oil-Spray Research at Penn State. By K. J. DeJuhasz.

MATERIAL

Corrosion and Heat-Resisting Steels as Applied to Automobile and Bus Use. By C. M. Johnson. Paper presented at the Detroit Regional Meeting of the American Society for Testing Materials, March 19, 1930. [G-1]

The author has been for a number of years concerned with researches on heat-resisting and corrosion resisting steels and has reported results of his investigations in a previous paper before the American Society for Steel Treating.

This paper gives a review of the past developments by briefly surveying the patent literature, and covers practical developments in stainless iron and steel. Silicon-chromium valve-steel and chromium-silicon nickel steels are considered, and data on their characteristics and results of corrosion tests are tabulated.

The Significant Properties of Automotive Lubricants. By H. C. Mougey. Paper presented at the Detroit regional meeting of the American Society for Testing Materials, March 19, 1930. [G-1]

Mr. Mougey discusses the evolution of attempts to specify the desirable properties of lubricating oils and points out that the tests such as gravity, color, flash-point, fire-point, emulsion test and so forth are not related to the lubricating value of a motor oil, but their use is in identifying oils, for controlling uniformity in production and for specifying and checking purchases. Thence he continues with a discussion of the qualities of oil from the standpoint of performance and the tests which bear a possible relation to these qualities.

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Notes and Reviews

Continued

The author lists the principal significant properties of automotive lubricants in the order of their importance, as follows: viscosity, stability, and oiliness, the last-named "under certain conditions." He warns that when motor oils are considered from the standpoint of performance in service the "mysterious thing called quality is not a definite property of an oil but is simply the result of using oils which are suitable for the uses to be made of them." He comments on the "excellent" work done by Committee D-2 of the A.S.T.M. in developing and standardizing oil tests, which, although not definite measures of quality, may be used to estimate quality in an oil for a particular use.

Automobile Bearing Metals. By Clair Upthegrove. Paper presented at the Detroit Regional Meeting of the American Society for Testing Materials, March 19, 1930. [G-1]

This paper points out some of the more dominant characteristics of metals used as bearing metals and some of the factors affecting those characteristics. The bearing metals are classed according to the principal base constituent, as tin base, lead base, copper or copper-tin base, zinc base and aluminum and magnesium base, and are discussed under the sub-heads of Babbitt Metal, Bronzes, and Light Metal-Base Alloys.

Studies in the Electrodeposition of Metals. By Donald B. Keyes and Sherlock Swann, Jr. Engineering Experiment Station Bulletin No. 206. Published by the University of Illinois, Urbana, Ill., May 1930; 20 pp. [G-1]

This study was undertaken with the hope that it might yield valuable results such as those which led to the discovery and widespread use of chromium-plated materials. Aluminum, beryllium, boron, chromium, tungsten, titanium, vanadium and cerium were studied. Chromium was included so as to determine whether it could be plated by a method other than from aqueous solution.

The results of the various tests are given in brief, and in conclusion the authors state that, since the continuous electrodeposition of the metals from aqueous solution or solvents of high dielectric constant failed in every case, they believe that the successful disposition of amphoteric metals must take place from highly ionized complex compounds. Aluminum was found to be the only metal that could be electrodeposited from the complex formed with tetraethyl ammonium bromide and its halide. Causes of failure of the other metals to plate are discussed and the intention is expressed to continue the investigation of the possibility of plating these metals from an organometallic compound in ether.

Metals Used in Aircraft Construction. By Bradley Stoughton. Published in *Metals and Alloys*, January, 1930, p. 317. [G-1]

By the use of numerous tables, the author, who is head of the Department of Metallurgical Engineering of Lehigh University, has presented a store of valuable data within a few pages. Typical strength-weight factors of aircraft materials; the chemical analysis, heat-treatment and physical properties of aluminum and its alloys; the same information on structural steels binary alloy steels, monel metal, carbon steels, aluminum-magnesium alloys and some of the more common heat-resisting alloys form the subjects of 10 tables included and explained in the article.

Phenol Resinoids and the Automobile. By L. V. Redman. Paper presented at the Detroit Regional Meeting of the American Society for Testing Materials, March 19, 1930. [G-4]

This paper discusses the applications of phenol resinoids in the automobile. Trade-marked by their inventor as Bakelite, these insulating materials are used throughout

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Notes and Reviews

Continued

the electrical system and as a cement for holding the bulbs in the headlamps and tail-lamp. Chief among the purely mechanical applications is the use of laminated phenol-resinoid canvas as a camshaft gear for silencing the operation of the timing train. In addition, a number of resinoid molded items are done in colors to conform to the color scheme of the car, such as shift-lever balls, control knobs, switch bases, dome-light frames, vanity vases and cigar lighters. Other indirect ways by which these materials have contributed to the development of the automotive industry are also mentioned, and a table giving the properties of fabricated phenol-resinoid products is included.

The uses of Bakelite and similar products in the construction of aircraft are discussed by John F. Hardecker in his article, The Aeronautical Uses of Bakelite and Similar Products, published in *Aviation*, Jan. 25, 1930, p. 144.

Evolution in Automobile Finishes. M. J. Callahan. Paper presented at the Detroit Regional Meeting of the American Society for Testing Materials, March 19, 1930. [G-5]

In introducing his subject, the author reviews the profound changes in materials and processes that have been demanded by the automobile industry in the last 10 years. The changes introduced in the field of automobile finishes have savored more of revolution than of evolution, he declares, stating that during that period the basic ingredient of body finishes has been changed to one of radically different chemical nature, that an entirely new technique of application has developed, that entirely new chemicals have been synthesized for use in these new finishes, and that this new art and change have resulted in great economies to the automotive manufacturer as well as in benefit of increased service and utility to the owner.

The paper covers body finishes, fender finishes, and engine and accessory finishes, and discusses the methods of application, with a brief survey of possible future trends.

Present-Day Methods in Production and Utilization of Automotive Cast Iron. By A. L. Boegehold. Paper presented at the Detroit Regional Meeting of the American Society for Testing Materials, March 19, 1930. [G-5]

The operation of the foundry of a modern automobile factory has been reduced insofar as possible to a scientific basis, with much of the old rule-of-thumb system eliminated as a result of study of the fundamental principles underlying each phase of the foundry business.

The preparation of suitable molds to receive and form the molten metal into useful shapes, and the delivery to the molds of molten cast iron of such composition and physical condition that it will form perfect castings possessing physical properties required to successfully meet the service for which they are intended, are the two main divisions of foundry activities that have been converted from an art to something more nearly approaching a science, the author points out.

This paper sets forth some of the measures used in a modern automobile-castings foundry for assuring a uniformly high grade product concurrent with economy of production and describes the irons used in automobiles.

MISCELLANEOUS

Effect of Repeated Daily Exposures of Several Hours to Small Amounts of Automobile Exhaust Gas. By R. R. Sayers, W. P. Yant, Edward Levy and W. B. Fulton. Public Health Bulletin No. 186. Published by the Treasury Department, United States Public Health Service, City of Washington, 1929; 58 pp., illustrated. [H-1]

The necessity of providing adequate ventilation to remove automobile exhaust gas was realized at the inception of the Holland Tunnel. The design of the ventilating system was based on fundamental engineering data obtained in a com-

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Notes and Reviews
Continued

prehensive investigation looking toward the safety and comfort of the general public passing through the tunnel. A limited investigation was made of the effects of daily exposure of long duration such as traffic officers or maintenance men might experience. However, the latter study was thought to be inadequate to the present situation. As a result, the New York State Bridge and Tunnel Commission and the New Jersey Interstate Bridge and Tunnel Commission continued the cooperation with the Bureau of Mines for further study, and the results are described in this report.

In summing up the results of this investigation, in which six men were exposed for 4 to 7 hr. daily over a period of 68 days to mixtures of gasoline-engine exhaust gas and air containing 2, 3, and 4 parts of carbon monoxide per 10,000 parts of air, the following are some of the conclusions drawn by the authors:

(1) Exposure to 2 parts carbon monoxide to 10,000 parts of air caused some subjects at rest or exercising mildly to experience slight but not discomforting symptoms in approximately 2 hr. and distinct frontal headaches of a discomforting nature in 3½ to 4 hr.

(2) Exposure to concentrations of 3 and 4 parts of carbon monoxide produced headaches in correspondingly shorter periods.

(3) Exercise, even though of a mild form, distinctly augmented the absorption of carbon monoxide and caused symptoms to appear after shorter exposure.

(4) The rate of elimination of carbon monoxide from the blood in a given period after exposure varied directly with the initial saturation, but exercise immediately after exposure markedly increased the speed of elimination.

(5) There were no apparent signs that the test produced effects deleterious to the health and physical well-being of the subjects. No lack of appetite, change of weight, or muscular weakness occurred.

(6) Psychological analysis employing tests that were thought would reveal any difference occurring in the performance of the subjects after exposure to carbon monoxide in the concentrations given, failed to show any distinct effect due to the gas, although there was a slight tendency for a poorer performance to be made on a prolonged steadiness test.

L'Automobile au Sahara. By Lieutenant-Colonel Gautsch. Published in *Omnia*, March, 1930, p. 634. **[H-1]**

How has the conquest of the Sahara by automobile been brought about? What are the characteristics of vehicles best suited to that region? Of what utility is automobile transportation there?

Answering these three questions, the author summarizes the almost epic struggle in which modern transportation engineering has been pitted against the discomforts, difficulties and dangers attendant upon travel over the desert wastes. Not only is the distance across the Sahara considerable, the shortest route being about 1200 miles, but the sources of water are widely scattered and even when reached afford often only a small supply of poor quality; the atmospheric temperature range is great and the wind and sand storms violent. Finally, perhaps the most serious obstacle from the automotive viewpoint is the great diversity in the consistency of the terrain, ranging from fine sand to hard rock.

A brief historical sketch is given of the various attempts made to cross the Sahara by automobile, but the author devotes most of his discussion to the technical aspects of the different enterprises, dwelling particularly on the expedients adopted to overcome the steep grades encountered, often greater than 20 per cent, the practically roadless stretches and the climatic conditions, in the face of which current engine-cooling methods are insufficient.

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Notes and Reviews

Concluded

La Danse des Milliards. By Pierre Maillard. Published in *La Vie Automobile*, Jan. 25, 1930, p. 33. **[H-3]**

The extent to which bicyclists and motorists contribute to the coffers of France is the theme, dwelt on rather acrimoniously, of this author. Tax receipts from automobiles have increased from 206,869,000 francs in 1924 to 686,237,000 in 1928, while the 8,000,000 cyclists of 1928 paid the government 119,252,000 francs, as compared with 35,466,000 francs contributed by their predecessors of four years previous.

However, the direct tax of 686,237,000 francs paid by autoists is but a drop in the bucket compared with the amount contributed indirectly by the industry, the total of which, in round figures, is about 3,000,000,000 francs. In short, for every vehicle of the average selling price of 26,000 francs, 5,500 francs is paid in wages to the workmen who made it and 2,600 francs to the government which watched them work. And, in spite of this heavy taxation, motorists' privileges are being curtailed; for instance, they may be deprived of the right of parking in certain streets of Paris.

What's Wrong with the Speedometer Anyway? By A. B. Creelman. *Bus Transportation*, January, 1930, p. 14. **[H-4]**

The author, who is superintendent of equipment for the Pennsylvania-Ohio Public Service, attributes to difficulty of maintenance the large number of speed-indicating instruments that are permanently out of order on motorcoaches. He describes various steps in the experimentation and development of an electrical speed-indicator by the Westinghouse company in cooperation with his own organization. One of these tachometers, which is mounted on Fageol motorcoach, has operated more than 150,000 miles without trouble. The magneto is driven by a coiled-wire belt from a pulley on the auxiliary shaft. The tachometer dial can be graduated either for miles per hour or engine revolutions, or both scales can be used on one dial. The writer expresses the opinion that provision for tachometer installation should be made either on the coach engine or on the transmission.

MOTOR COACH AND MOTOR-TRUCK

Organisation und Einrichtungen der Verkraftung der Deutschen Reichspost. Published in *Automobiltechnische Zeitschrift*, March 10, 1930, p. 167. **[J and K-2]**

Since 1926 the number of vehicles in the service of the German postal system, which includes the transportation of passengers as well as the delivery of mail and telegrams, has increased as follows: motorcoaches, from 2159 to 3728; passenger-cars, from 201 to 409; three-wheel delivery cars, from 187 to 861 and four-wheel delivery cars from 238 to 1317.

Most of the present article is devoted to a description of the service organization engaged in attending to the maintenance and repairing of these vehicles. The most important unit of the organization is the Berlin-Borsigwalde garage, employing about 600 persons. To this shop is assigned the especial duty of oversight of the construction of vehicles; the purchase of fuels, lubricants and other materials; and the testing and research incidental to this work. Four other large garages employ about 200 persons each and devote their energies solely to major repairs and the thorough overhauling that comes after a minimum of about 35,000 miles have been run. Subordinate to these garages are shops in which from 20 to 30 men are employed in making minor repairs and installing units that have been conditioned and assembled in the main garages. Still lower in the scale are maintenance stations to remedy minor troubles without loss of vehicle time. Finally, one-man service stations are engaged in preventive maintenance work.

A further installment of this article will trace the part of experience in selecting and perfecting vehicles for the postal service and describe the Berlin-Boriswalde garage.

Personal Notes of the Members

Continued

Max P. Baker has left his position with the Oakland Motor Car Co., of Pontiac, Mich., and is now an engineer with the Waco Aircraft Co., Inc., of Troy, Ohio.

Walter C. Becker has been advanced from the position of superintendent of bus service and assistant to the superintendent of the utility department to the position of automotive engineer. He will supervise and be responsible for the coordination of all trolley-bus, motorcoach and other automotive activities for the Chicago Surface Lines.

Vincent Bendix, president of the Bendix Aviation Corp., was the guest of honor at a testimonial banquet given by the American Sons and Daughters of Sweden at the Stevens Hotel, Chicago, on April 23. The banquet was given in recognition of the distinction recently conferred upon Mr. Bendix when he was decorated as commander of the Royal Order of the North Star by King Gustav at Stockholm, Sweden.

Clark H. Brown has accepted a position with the Denney Publishing Co., of New York City, and will be engaged in sales development work. Until lately he was automotive engineer with the Texas Co., at Norfolk, Va.

Lowell G. Brown recently was elected president of the Weymann Holding Co., of New York City.

R. C. Callahan has relinquished his post with the Oakland Motor Car Co., of Pontiac, Mich., and is now an engineer with the Stromberg Carburetor Co., of Detroit.

Ralph L. Corey recently assumed the duties of vice-president and general manager of the Van Sicklen Corp., of Elgin, Ill. His previous connection had been with the National Gauge Equipment Co., of La Crosse, Wis., which he served as sales manager.

Herbert G. Fales, former industrial engineer with E. I. duPont de Nemours & Co., of Wilmington, Del., is now assistant to the vice-president of the International Nickel Co., in New York City.

Wayne H. Farnsworth, until recently experimental engineer at the Berkeley, Cal., plant of the Hall-Scott Motor Car Co., has been transferred to New York City, where he is engaged in marine sales and service work.

F. J. Griffiths, former chairman of the board of the Alloy Steel Corp., will head the new research unit of the Republic Research Corp., with headquarters at Massillon, Ohio.

William R. Grundmann has been transferred from Los Angeles to Kansas City, Mo., where he is engaged in performing the duties of automotive engineer for the national sales department of the Texas Co.

T. W. Hallerberg has accepted a position as designer with the Pickwick Motor Coach Works, of Inglewood, Calif. Prior to making this connection he was chief engineer of Lubrication Devices, Inc., of Battle Creek, Mich.

Frank B. Hanford, until lately in the independent field as a consulting engineer, is now in the sales division of Beers & De Long, of New York City.

Howe H. Hopkins has been appointed service engineer for the Godward Gas Generator Co., Inc., of New York City. His previous connection was with the Bijur Lubricating Corp., also of New York City, as engineer.

Sidney L. Jacoby, a former New York University student, recently entered the service of the Amtorg Trading Corp., of New York City, as a junior draftsman.

S. Johnson, Jr., who has been serving the Westinghouse Air Brake Co., of Pittsburgh, as general engineer in the automotive brake division, has recently been made chief engineer of the Bendix-Westinghouse Air Brake Co., also of Pittsburgh.

Harry D. Kinnear, who is serving the Gabriel Snubber & Mfg. Co., of Cleveland, was recently appointed Pacific Coast representative of that company.

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Personal Notes of the Members

Continued

Ralph S. Lane has resigned as president and general manager of United Motors Service, Inc., of Detroit, and accepted a special assignment on the central staff of the General Motors Corp., in Detroit.

Eric Langlands is now chief engineer and general manager of the Continental Coach Corp., of the city of Washington. He previously held a similar position in the Cleveland engineering office of the Majestic Car Corp.

Charles H. Lindsay, until recently resident engineer with the American LaFrance & Foamite Corp., Inc., at Cleveland, has been transferred and is now engaged in industrial development work at the company's plant in Elmira, N. Y.

Edward B. McCune has resigned as designing engineer for the Chrysler Corp., of Detroit, and has joined the engineering staff of the commercial car division of the Studebaker Corp., of South Bend, Ind.

E. B. Middlekauf, who was serving the Texas Co. in New York City as secretary to the manager of purchasing, has been promoted to the duties of purchasing agent.

John M. Miller, former president of the New Brunswick Airport, at New Brunswick, N. J., is now field manager for the Westchester Airport Corp., of Armonk, N. Y.

J. P. Miller is now a designer with Duesenberg Brothers, of Indianapolis. He was previously a design draftsman for the Horace E. Dodge Boat Works, of Detroit.

Frank C. Mock, engineer with the Bendix Aviation Corp., of South Bend, Ind., has been transferred to the Eclipse Aviation Corp., division of the Bendix Aviation Corp., at East Orange, N. J.

R. L. Morrison, until lately representative of the Westinghouse Air Brake Co., at Wilmerding, Pa., has been given the position of Detroit district sales manager with the new Bendix-Westinghouse Automotive Air Brake Co.

A. C. Munning, formerly research director of the Hanson-Van Winkle-Munning Co., of Matawan, N. J., is now vice-president in charge of engineering of Munning & Munning, of Philadelphia.

Charles E. Nelsen, former service superintendent of the Harrington Motor Co., of Minneapolis, is now occupying a similar position with the Northern Motor Co., also of Minneapolis.

John Pollitt, Jr., formerly director of Pollitt & Son., Ltd., of Liverpool, England, is now proprietor of Pollitt & King, also of Liverpool.

Lewis Morgan Porter, formerly a member of the engineering department, aircraft division, of the Lycoming Mfg. Co., of Williamsport, Pa., is now research engineer in the automotive laboratory of the Vacuum Oil Co., at Paulsboro, N. J.

Harry J. Richards, now chief engineer of the Atterbury Motor Car Co., of Buffalo, recently assumed his position after leaving the Larrabee-Deyo Motor Truck Co., of Binghamton, N. Y., of which he was vice-president and chief engineer.

Announcement has been made of the appointment of **J. L. Rupp**, former president of the Wubco Battery Corp., of Swissvale, Pa., to the post of vice-president in charge of engineering and manufacture for the Gould Storage Battery Co., Inc., of Depew, N. Y.

Frederic Saltzman is now a member of the engineering and planning press division of the General Electric Co., of Lynn, Mass. His former connection was with the W. F. Stewart Co., of Flint, Mich., by which he was employed as a die designer.

B. J. Smith, formerly manager of the national accounts division for the Autocar Co., of Ardmore, Pa., lately assumed the duties of manager of national account sales with the Goheen Corp. of New Jersey. His headquarters will be in New York City.

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Personal Notes of the Members

Concluded

E. G. Soash has accepted a position as mechanical engineer in development work with the DuPont Ammonia Corp., of Belle, W. Va. He was formerly in the sales division of the Toledo Machine & Tool Co., of Toledo, Ohio.

Charles F. Stein, former tool designer with the Nash Motors Co., of Kenosha, Wis., is now occupying a similar position with the Waukesha Motors Co., of Waukesha, Wis.

H. M. Stillman has accepted a position as sales engineer for the Saginaw Steering Gear Division of the General Motors Corp., at Saginaw, Mich. He previously was Detroit branch manager of the North East Electric Co., of Rochester, N. Y.

Max W. Thaete, former service manager of the Jackson Chevrolet Co., of Pueblo, Col., is now employed as distributor of mechanical tools and equipment by the Truth Tool Co., of Mankato, Minn.

S. B. Tompkins, who was previously service manager for the Mack-International Motor Truck Corp., of Chicago, is now associated with Fred L. Meckel, also of Chicago, designer and manufacturer of automobile bodies.

Harvey E. Tyler has severed his connection with the Stewart-Warner Corp., of Chicago, which he was serving as research engineer, to accept a similar position with the Weaver Mfg. Co., of Springfield, Ill.

Ch. A. Viriot, former vice-president of Silentbloc, Inc., of Detroit, sailed recently on the S. S. France for Paris, where it is understood he will reside for a time. His plans for the future were unannounced.

Ernest Kurt von Brand is now an associate editor of the Engineering Index of the Society of Mechanical Engineers in New York City. Prior to assuming this post he was research engineer in the dynamics section of the General Motors Corp. Research Laboratories, in Detroit.

S. A. E. Journal

JANUARY, 1930

REPORTS OF DIVISIONS TO STANDARDS COMMITTEE

THE SOCIETY OF AUTOMOTIVE ENGINEERS, INC.
29 WEST THIRTY-NINTH STREET
NEW YORK CITY

Reports of Divisions to Standards Committee

Standards Committee Meeting Jan. 20

Crystal Ball Room—Book-Cadillac Hotel—Detroit, Mich.

IN this pamphlet are printed reports that have been prepared for submission to the Standards Committee and to the Society by 12 Divisions of the Standards Committee since the Semi-Annual Meeting last June.

All of the reports are submitted at this time for approval after having been considered carefully by the respective Divisions and given as wide publicity as possible by publication in the S.A.E. JOURNAL from month to month. The reports as now presented are believed to be in acceptable form, and any proposals should be only in the nature of important and carefully considered constructive changes.

Under the Standards Committee procedure, these reports may be approved as presented, amended within limitations or referred back to the respective Divisions for sufficient reason. The action taken on them by the Standards Committee will be passed upon by the Council and the general business session of the Society and those approved will be published in the S.A.E. HANDBOOK.

Rejection or major changes in any of the reports will require that they be sent back to the Divisions that prepared them and that they cannot be passed upon again before the Semi-Annual Meeting of the Society next June. In voting on the reports at the Standards Committee Meeting, the Regulations require that only members of the Standards Committee do so.

Agricultural Power Equipment Division

PERSONNEL

O. B. Zimmerman, *Chairman*	International Harvester Co.
R. O. Hendrickson, *Vice-Chairman*	J. I. Case Threshing Machine Co., Inc.
A. H. Gilbert	Rock Island Plow Co.
P. E. Holt	Caterpillar Tractor Co.
H. E. McCray	John Deere Tractor Co.
John Mainland	Advance-Rumely Co.
R. L. Miller	Hart Grain Weigher Co.
A. C. Rasmussen	Insley Mfg. Co.
O. W. Sjogren	Killefer Mfg. Corp.
G. A. Young	Purdue University

Transmissions, Roller Chains, Sprockets and Cutters

(*Proposed American Standard*)

Roller-chain standardization was initiated in 1917 when definite recommendations for the pitches, roller diameters, pin diameters, side-plate thickness and chain widths were adopted. This standard was expanded and, together with specifications for sprockets an sprocket cutters, was adopted in 1921. Study of the standard continued jointly with a Committee of the American Society of Mechanical Engineers until December, 1924, when the Sectional Committee on Transmission Chains, Sprockets and Cutters was organized under the procedure of the American Standards Association, then the American Engineering Standards Committee, to review the standards then existing and to formulate a report for adoption as American Standard that would be applicable to uses other than automotive drives.

The Sectional Committee represented practically all of the roller-chain manufacturers and also other interests concerned with roller-chain application. The sub-committee on roller chain that was organized has made a careful study of roller chains, sprockets and cutters and their use in current practice. Contacts were maintained with roller-chain interests abroad to secure, so far as possible, international uniformity in roller-chain standardization. The report of the Roller Chain Sub-Committee has been approved by the Sectional Committee and referred to the sponsors of the Sectional Committee, which are the Society, the American Society of Mechanical Engineers and the American Gear Manufacturers Association, for their approval and for submission to the American Standards Association for final approval by that body so far as representativeness of the Sectional Committee and its procedure are concerned.

The report having been submitted to the Society, it was assigned by the Council to the Agricultural Power Equipment Division of the Standards Committee for recommendation to the Standards Committee and the Society for approval. The Division now submits the report for such approval and submission to the American Standards Association for final approval and adoption as American Standard. No part of the report is submitted at this time for adoption as S.A.E. Standard pending its final approval, following which it will be submitted to the Standards Committee as the basis for revision or other recommendation concerning the present S.A.E. Standard on Roller Chains commencing on p. 89 of the 1929 edition of the S.A.E. HANDBOOK.

ROLLER CHAIN SUB-COMMITTEE REPORT

The roller chain to which this report relates is the type commonly used in power transmissions on motor-trucks, motorcycles, tractors, industrial machinery, machine tools and similar applications. The sizes and dimensions specified are for standard chains that are considered suitable for all usual chain applications. Adoption of the standard does not indicate that other chains will not be obtainable for use where they are required. It is recommended by the Sectional Committee, that the chain manufacturers be consulted before final selection of chain is made by users inasmuch as this report is subject to periodic revisions to conform with developments in the arts.

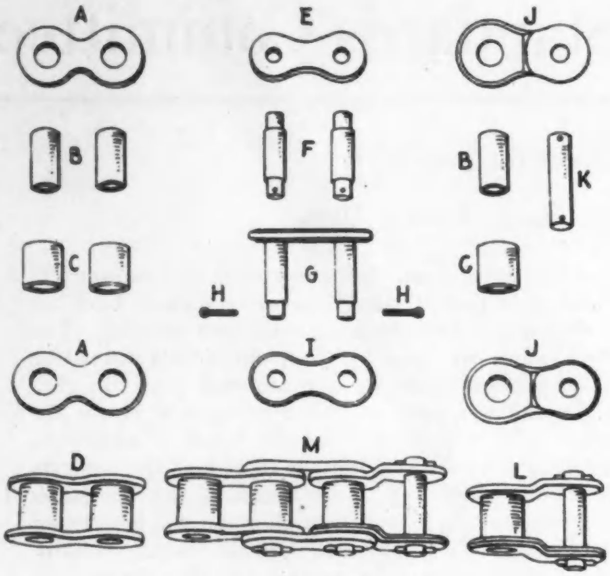


FIG. 1—ROLLER CHAIN PARTS

Roller Chain Parts Nomenclature

Roller Link D.—An inside link consisting of two inside plates, two bushings, and two rollers.

Pin Link G and E.—An outside link consisting of two pin-link plates assembled with two pins.

Inside Plate A.—One of the plates forming the tension members of a roller link.

Pin-Link Plate E.—One of the plates forming the tension members of a pin link.

Pin F.—A stud articulating within a bushing of an inside link and secured at its ends by the pin-link plates.

Bushing B.—A cylindrical bearing in which the pin turns.

Roller C.—A ring or thimble which turns over a bushing.

Assembled Pins G.—Two pins assembled with one pin-link plate.

Connecting-Link G and I.—A pin link having one side plate detachable.

Connecting-Link Plate I.—The detachable pin-link plate belonging to a connecting link.

Offset Link L.—A link consisting of two offset plates assembled with a bushing and roller at one end and an offset-link pin at the other.

Offset Plate J.—One of the plates forming the tension members of the offset link.

Offset Link Pin K.—A pin used in offset links.

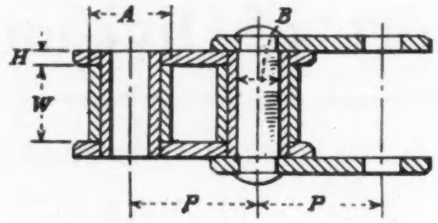


FIG. 2—LINK ASSEMBLY

Roller Diameters are approximately 5/8 *P*.

Widths.—The width is defined as the minimum distance between the inside plates. In the wide series the width is the nearest binary fraction to 5/8 *P*. In the narrow series it is the nearest binary fraction to 0.41 *P*. The narrow chains are seldom used and are not recommended except for unusual cases.

Pin Diameters are approximately 5/16 *P* or 1/2 of the roller diameter.

Thickness of Inside Plates for the standard series is approximately 1/8 *P*.

Test Load.—This is the load under which a chain should be tested for defects of material. For single-width chains it is calculated from the formula

$$42000\ (\text{Pin Diameter})^2 - 280\ \text{lb.} \qquad (1)$$

Test loads for multiple-width chains will be proportionately greater than those for single-width chains.

Measuring Load.—This is the load under which a chain should be measured for length. It is equal to the standard test-load divided by 32.

Standard Chain Numbers.—The right-hand figure in the chain number is zero for roller chains of the usual proportions, 1 for a light-weight chain and 5 for a rollerless bushing chain. The numbers to the left of the right-hand figure denote the number of 1/8 inches in the pitch. The letter *H* following the chain number denotes the extra-heavy series. The letter *D* prefixed to the chain number denotes a double-width, *E* a triple-width, *F* a quadruple-width chain and so forth.

Extra-Heavy Series.—These chains have thicker inside plates than those of the regular standard. The rollers, bushing diameters, pin diameters, widths, test loads and measuring loads are the same as in the standard series. The chain numbers are those of the standard series with the letter *H* appended. Thus the number 80*H* denotes a 1-in. pitch extra-heavy chain with inside plates 0.156 in. thick.

TABLE 1—GENERAL CHAIN DIMENSIONS

		Standard Series						Extra Heavy Series
Pitch *P*	Roller Diameter *A*	Width *W*	Pin Diameter *B*	Thickness of Inside Plate *H*	Test Load, Lb.	Measuring Load, Lb.	Standard Chain No.	Thickness of Inside Plate *H*
3/8	0.200	3/16	0.141	0.050	555	17	35N	.....
1/2	5/16	5/16	0.156	0.060	740	23	40	.....
5/8	0.400	3/8	0.200	0.080	1,400	44	50	.....
3/4	15/32	1/2	0.234	0.094	2,018	63	60	0.125
1	5/8	5/8	0.312	0.125	3,820	119	80	0.156
1 1/4	3/4	3/4	0.375	0.156	5,626	176	100	0.187
1 1/2	7/8	1	0.437	0.187	7,760	242	120	0.219
1 3/4	1	1	0.500	0.219	10,220	319	140	0.250
2	1 1/8	1 1/4	0.562	0.250	13,008	406	160	0.281
2 1/2	1 9/16	1 1/2	0.781	0.312	25,340	792	200	0.375
3	1.90	1 7/8	0.937	0.375	36,634	1,145	240	0.500

TABLE 2—ULTIMATE DIMENSIONAL LIMITS FOR INTERCHANGEABLE CHAIN-LINKS

	Chain Pitches, In.									
	3/8	1/2	5/8	3/4	1	1 1/4	1 1/2	1 3/4	2	2 1/2
Minimum Distance between Inside Plates........	0.188	0.313	0.375	0.500	0.625	0.750	1.000	1.000	1.250	1.500
Maximum Width of Roller Links, Standard Series	0.295	0.440	0.544	0.697	0.886	1.075	1.391	1.457	1.770	2.147
Maximum Width of Roller Links, Extra-Heavy Series........	....	....	....	0.761	0.950	1.142	1.456	1.519	1.832	2.275
Minimum Distance between Pin Plates, Standard Series........	0.297	0.442	0.546	0.699	0.888	1.077	1.393	1.459	1.772	2.149
Minimum Distance between Pin Plates, Extra-Heavy Series........	....	....	....	0.763	0.952	1.144	1.458	1.521	1.834	2.277
Maximum Pin-Diameter........	0.1415	0.1565	0.2005	0.2345	0.313	0.3755	0.438	0.5005	0.563	0.782
Minimum Hole in Bushing........	0.1425	0.1575	0.202	0.236	0.315	0.378	0.4405	0.5035	0.5665	0.786
Minimum Clearance between Pins and Bushings	0.001	0.001	0.0015	0.0015	0.002	0.0025	0.0025	0.003	0.0035	0.004
Minimum Value of x for Offset Plates........	0.168	0.220	0.271	0.322	0.425	0.527	0.630	0.732	0.835	1.040
Minimum Value of y for Offset Plates........	0.193	0.252	0.312	0.371	0.490	0.609	0.727	0.846	0.965	1.202

Light-Weight Machinery Chain.—This chain is designated as No. 41. It is ½-in P; ¼ in. wide; has 0.306-in. diameter rollers; 0.141-in. pin diameter; and 0.050-in. thick side plates. The test load is 555 lb. and the measuring load is 17 lb.

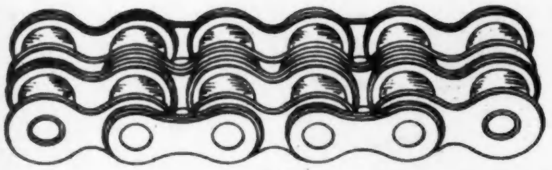


FIG. 3—DOUBLE-WIDTH ROLLER CHAIN

Multiple-Width Chains.—The standard thickness of center plates in these chains is equivalent to two thicknesses of the inside plates as used on the roller links.

Tolerances for Chain Length.—New chains under standard measuring-load are allowed to run over-length 1/64 in. per ft., but must not be under-length.

Tolerances for Chain Parts.—To insure interchangeability between connecting links as produced by different makers of chain the following standard maximum and minimum tolerances are adopted for the guidance of the manufacturers. They are not the actual tolerances to be used in manufacturing but rather the limiting tolerances, maximum and minimum, within which it is necessary to keep to insure the desired interchangeability.

The minimum distance between inside plates of roller links is equal to the nominal width of chain.

Maximum pin-diameters to be nominal + 0.0005 in. (2)

Minimum hole in bushing to be 1.006 nominal pin-diameter + 0.0005 in. (3)

Minimum clearance between pins and bushings would then be 0.006 nominal pin-diameter. (4)

Maximum width of roller link to be nominal width of chain + 2.05 nominal side plate thickness + C (5)

where

C = 0.005 in. for side plates up to and including 0.156 in. thick and 0.007 in. for side plates over 0.156 in. thick.

Minimum distance between pin plates to be maximum width of roller link + 0.002 in. (6)

Standard offset links are to be made so as to accommodate chains having inside plates with a maximum width, after being beveled, equal to 95 per cent of the pitch, and pin plates with a maximum width, after being beveled, equal to 82 per cent of the pitch. To accomplish this it is necessary that the standard minimum values of x and y (Fig. 4) shall be:

$$x_{min} = 0.41\,P + 0.015 \text{ in.} \qquad (7)$$

$$y_{min} = 0.475\,P + 0.015 \text{ in.} \qquad (8)$$

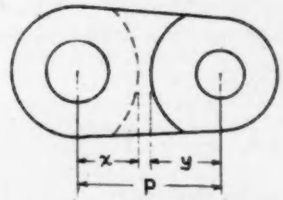
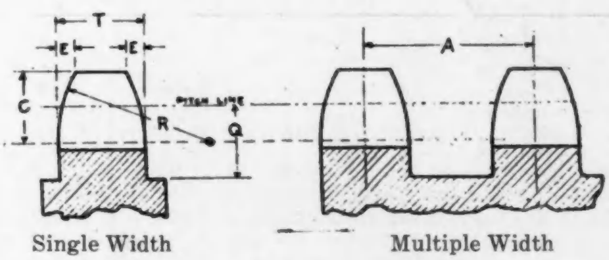


FIG. 4—OFFSET PLATE

Sprockets for Roller Chains



Single Width Multiple Width

FIG. 5—SPROCKET TOOTH SECTIONS

The section profile shows the recommended chamfering of sprocket teeth for roller chains

P = Pitch of chain
W = Chain width
$T = 0.93\,W - 0.006$ in. (for single-width chains)
$T = 0.90\,W - 0.006$ in. (for double and triple-width chains)
$T = 0.88\,W - 0.006$ in. (for quadruple-width chains and over)
$C = 0.5\,P$

TABLE 3—SPROCKET THICKNESS

Width of Chain W	Nominal Sprocket Thickness T	Tolerances on Thickness + or −	Nominal Clearance $W - T$
3/16	0.168	0.006	0.019
1/4	0.227	0.007	0.023
5/16	0.284	0.008	0.028
3/8	0.343	0.009	0.032
1/2	0.459	0.012	0.041
5/8	0.575	0.014	0.050
3/4	0.692	0.017	0.058
1	0.924	0.022	0.076
1 1/4	1.156	0.027	0.094
1 1/2	1.389	0.032	0.111

TABLE 4—SPROCKET CHAMFER

Pitch P	Depth of Chamfer C	Width of Chamfer E	Minimum Radius R
3/8	3/16	3/64	0.398
1/2	1/4	1/16	0.531
5/8	5/16	5/64	0.664
3/4	3/8	3/32	0.796
1	1/2	1/8	1.062
1 1/4	5/8	5/32	1.327
1 1/2	3/4	3/16	1.593
1 3/4	7/8	7/32	1.858
2	1	1/4	2.124
2 1/2	1 1/4	5/16	2.654

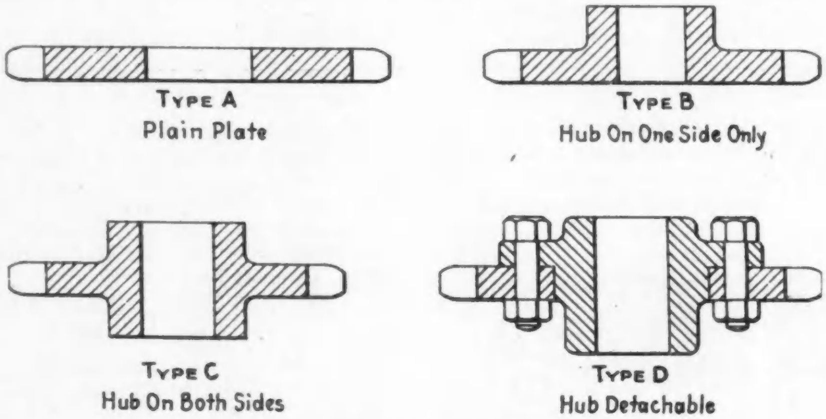


FIG. 6—TYPES OF ROLLER-CHAIN SPROCKETS

$E = 1/8\,P$
$R_{min} = 1.063\,P$
$Q = 0.5\,P$
$A = W + 4.15H + 0.003$ in.
H = Nominal thickness of inside plate

Tolerance on Sprocket Thickness $= \pm\,[(W/50) + 0.002$ in.$]$ (9)

Pitch Diameter of Sprocket $= P/\sin\,(180$ deg.$/N)$ (10)

where
P is the pitch and N is the number of teeth.

TABLE 5—NEGATIVE TOLERANCES ON THE BOTTOM DIAMETERS OF CUT SPROCKETS FOR VARIOUS NUMBERS OF TEETH

Pitch P	Number of Teeth T									
	9	16	25	36	49	64	81	100	121	144
3/8	0.004	0.004	0.004	0.005	0.005	0.006	0.006	0.006	0.007	0.007
1/2	0.004	0.005	0.0055	0.006	0.0065	0.007	0.0075	0.008	0.0085	0.009
5/8	0.005	0.0055	0.006	0.007	0.008	0.009	0.009	0.009	0.010	0.011
3/4	0.005	0.006	0.007	0.008	0.009	0.010	0.010	0.011	0.012	0.013
1	0.006	0.007	0.008	0.009	0.010	0.011	0.012	0.013	0.014	0.015
1 1/4	0.007	0.008	0.009	0.010	0.012	0.013	0.014	0.016	0.017	0.018
1 1/2	0.007	0.009	0.0105	0.012	0.013	0.015	0.016	0.018	0.019	0.021
1 3/4	0.008	0.010	0.012	0.013	0.015	0.017	0.019	0.020	0.022	0.024
2	0.009	0.011	0.013	0.015	0.017	0.019	0.021	0.023	0.025	0.027
2 1/2	0.010	0.013	0.015	0.018	0.020	0.023	0.025	0.028	0.030	0.033

Minimum Outside Diameter of Sprocket =
$P\,[0.6 + \cot\,(180\text{ deg.}/N)]$ (11)

Bottom Diameter of Sprocket =
Pitch Diameter − Roller Diameter (12)

Maximum Hub Diameter of Sprockets =
Pitch Diameter − Pitch (13)

Tolerances on bottom diameters of cut sprockets;
Positive tolerance = 0.000 in.
Negative tolerance = 0.003 in. + $0.001\,P\sqrt{N}$ (14)

Types of Sprockets.—For the designations of the four principal types of sprocket refer to Fig. 6.

Design of Standard Sprocket Teeth

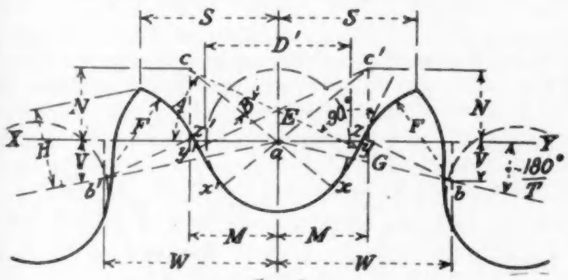


FIG. 7—DESIGN OF STANDARD SPROCKET TEETH

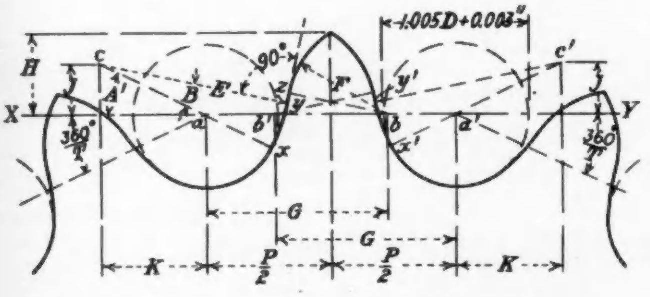


FIG. 8—DESIGN OF STANDARD SPROCKET TEETH

P = Pitch
D = Nominal roller diameter
T = Number of teeth
$D' = 1.005\,D + 0.003$ in.
$A = 35\text{ deg.} + (60/T)$
$A' = 35\text{ deg.} - (120/T)$
$B = 18\text{ deg.} - (56/T)$
$ac = 0.8\,D$
$M = 0.8\,D\cos\,[35\text{ deg.} + (60/T)]$
$N = 0.8\,D\sin\,[35\text{ deg.} + (60/T)]$
$J = 0.8\,D\sin\,[35\text{ deg.} - (120/T)]$
$K = 0.8\,D\cos\,[35\text{ deg.} - (120/T)]$
$E = 1.3025\,D + 0.0015$ in.
Chord $xy = (2.605\,D + 0.003)\sin\,[9\text{ deg.} - (28/T)]$

$$yz = D\left[1.24\sin\left(17\text{ deg.} - \frac{64}{T}\right) - 0.8\sin\left(18\text{ deg.} - \frac{56}{T}\right)\right]$$

$G = 1.24\,D$
$W = 1.24\,D\cos\,(180\text{ deg.}/T)$
$V = 1.24\,D\sin\,(180\text{ deg.}/T)$

$$F = D\left[0.8\cos\left(18\text{ deg.} - \frac{56}{T}\right) + 1.24\cos\left(17\text{ deg.} - \frac{64}{T}\right) - 1.3025\right] - 0.0015\text{ in.}$$

$$H = \sqrt{F^2 - [1.24\,D - (P/2)]^2}$$

When $1.24\,D$ is less than $P/2$, then $H = F$
$S = (P/2)\cos\,(180\text{ deg.}/T) + H\sin\,(180\text{ deg.}/T)$

Outside diameter of sprocket when tooth is pointed =
$P\cot\,(180\text{ deg.}/T) + 2\,H$

Minimum outside diameter of blank =
$P\,[0.6 + \cot\,(180\text{ deg.}/T)]$

The pressure angle for a new chain xab =
$35\text{ deg.} - (120/T)$

The minimum pressure angle $xab - B$ =
$17\text{ deg.} - (64/T)$

The average pressure angle = $26\text{ deg.} - (92/T)$

Four types of sprocket cutters are used, namely,

Space Cutters, of which five will be required to cut from 7 teeth up for any given roller diameter. The ranges are respectively 7-8, 9-11, 12-17, 18-34 and 35 teeth and over. The use of less than 7 teeth is discouraged, but when necessary, single-purpose cutters of this type are to be used.

Straddle Cutters, of which two will be required to cut from 7 teeth up for any given pitch and roller diameter. Cutter B is recommended for 17 teeth and under or for more than 17 teeth if a low pressure angle is desired. Cutter A is recommended for 18 teeth and over or for less than 18 teeth if a large pressure angle is desired and the arc of contact between chain and sprocket is fairly large.

Hobs, of which only one will be required to cut any number of teeth for a given pitch and roller diameter.

Fellows Cutters, for use on the Fellows gear shaper, of which not more than two will be required to cut any number of teeth for a given pitch and roller diameter.

Design of Standard Space-Cutters for Sprockets

Space cutters are made for the following ranges of teeth: 7-8, 9-11, 12-17, 18-34 and 35 and over. The lowest number of teeth in any group is designated by n and the highest by N.

The cutters are based on an intermediate number of teeth T, equal to $2\,Nn/(N + n)$ but the topping curve zs is based on N teeth. The values of T for the respective cutters are 7.47; 9.9; 14.07; 23.54 and 56.

Space cutters designed for a given roller diameter D will cut sprockets of any pitch.

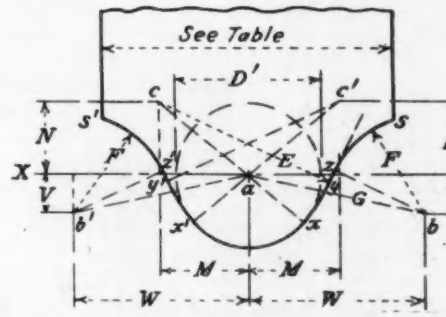


FIG. 9—SPACE CUTTER

Construction.—Referring to Fig. 9, draw XY. With a as center and a radius ax equal to $D'/2$ draw the circular arc for the seating curve xx' (Table 7). Locate c and c' from dimensions M and N (Table 6). With c and c' as centers describe the arcs xy and $x'y'$. Draw yz perpendicular to cy. Locate b from dimensions W and V, draw bz parallel to cy, and with radius bz, equal to F (Table 6), draw the topping curve zs. The line yz is a common tangent to the two circular arcs xy and zs.

Supplementary Data.—Tables 6, 7 and 8 give all necessary data for the construction of standard space-cutters. Supplementary formulas used in their calculation are given below.

The angle Yab is 180 deg./T when the cutter is made for

TABLE 6—DATA FOR LAYING OUT CUTTER OUTLINES

Range of Teeth	M	N	W	V	F	Chord xy	yz
7–8	$0.585D$	$0.5457D$	$1.1327D$	$0.504D$	$0.7104D-0.0015$	$0.2384D+0.0003$	$0.039D$
9–11	$0.599D$	$0.5303D$	$1.1782D$	$0.387D$	$0.6981D-0.0015$	$0.28D+0.0003$	$0.056D$
12–17	$0.619D$	$0.5068D$	$1.2128D$	$0.258D$	$0.6807D-0.0015$	$0.3181D+0.0004$	$0.090D$
18–34	$0.634D$	$0.4879D$	$1.2353D$	$0.108D$	$0.6542D-0.0015$	$0.354D+0.0004$	$0.146D$
35 up	$0.647D$	$0.471D$	$1.24D$	0	$0.6345D-0.0015$	$0.385D+0.0004$	$0.171D$

TABLE 7—SEATING-CURVE DIAMETERS

Roller Diameter D	Seating-Curve Diameter D'	Roller Diameter D	Seating-Curve Diameter D'
0.200	0.204	5/8	0.631
1/4	0.254	3/4	0.757
0.306 and 5/16	0.317	7/8	0.882
0.400	0.405	1	1.008
15/32	0.474	1 1/8	1.134
9/16	0.568	1 9/16	1.574

T teeth only, but it has values as given in Table 8 for cutters covering a range of teeth as here designed. Accordingly the following formulas are special for cutters covering the standard ranges of teeth,

$$W = 1.24\,D \cos Yab$$
$$V = 1.24\,D \sin Yab$$
$$bc' = b'c = \sqrt{[(W - M)^2 + (V + N)^2]}$$
$$yz = D\left[1.24 \sin\left(17 \text{ deg.} + \frac{116}{T} - Yab\right) - 0.8 \sin\left(18 \text{ deg.} - \frac{56}{T}\right)\right]$$

TABLE 8—CHECKS

Number of Teeth	E	G	$b - c$ $b' - c'$	$b - c'$ $b' - c$	Angle Yab
7–8	$1.3025D + 0.0015$	$1.24D$	$2.0133D$	$1.1841D$	24 deg.
9–11	$1.3025D + 0.0015$	$1.24D$	$2.0014D$	$1.0847D$	18 deg. 10 min.
12–17	$1.3025D + 0.0015$	$1.24D$	$1.9852D$	$0.9682D$	12 deg.
18–34	$1.3025D + 0.0015$	$1.24D$	$1.9621D$	$0.8465D$	5 deg.
35 up	$1.3025D + 0.0015$	$1.24D$	$1.9445D$	$0.7576D$	0 deg.

TABLE 9—RECOMMENDED CUTTER SIZES FOR ROLLER-CHAIN SPROCKETS

		For Number of Teeth T												
		Cutter Diameter (Minimum)						Cutter Width (Minimum)						
Pitch	Roller Diameter	6	7–8	9–11	12–17	18–34	35 and Over	6	7–8	9–11	12–17	18–34	35 and Over	Bore
*3/8	0.200	2 3/4	2 3/4	2 3/4	2 3/4	2 3/4	2 3/4	15/32	15/32	15/32	7/16	7/16	13/32	1
3/8	0.250	2 7/8	2 7/8	2 7/8	2 7/8	2 7/8	2 7/8	15/32	15/32	15/32	7/16	7/16	13/32	1
3/8 to 1/2	0.250	2 3/4	2 7/8	2 7/8	2 7/8	2 7/8	2 7/8	5/8	5/8	19/32	19/32	9/16	9/16	1
1/2	0.313	2 7/8	3	3	3	3	3	5/8	5/8	19/32	19/32	9/16	9/16	1
*1/2 to 5/8	0.313	3	3	3 1/8	3 1/8	3 1/8	3 1/8	3/4	3/4	3/4	3/4	23/32	11/16	1
*5/8	0.400	3 1/8	3 1/8	3 1/4	3 1/4	3 1/4	3 1/4	3/4	3/4	3/4	3/4	23/32	11/16	1
5/8 to 3/4	0.400	3 1/4	3 1/4	3 3/8	3 3/8	3 3/8	3 3/8	29/32	29/32	29/32	7/8	27/32	13/32	1
*3/4	0.469	3 1/4	3 1/4	3 3/8	3 3/8	3 3/8	3 3/8	29/32	29/32	29/32	7/8	27/32	13/16	1
3/4 to 1	0.469	3 1/4	3 3/8	3 3/8	3 1/2	3 1/2	3 1/2	1 1/4	1 1/4	1 3/16	1 5/32	1 1/8	1 3/32	1
*1	0.563	3 3/4	3 7/8	3 7/8	4	4	4	1 1/4	1 1/4	1 3/16	1 5/32	1 1/8	1 3/32	1 1/4
1	0.625	3 7/8	4	4	4 1/8	4 1/8	4 1/8	1 1/4	1 1/4	1 3/16	1 5/32	1 1/8	1 3/32	1 1/4
*1 to 1 1/4	0.625	3 7/8	4	4 1/8	4 1/8	4 1/4	4 1/4	1 1/2	1 1/2	1 15/32	1 15/32	1 13/32	1 11/32	1 1/4
1 1/4	0.750	4	4 1/8	4 1/4	4 1/4	4 3/8	4 3/8	1 1/2	1 1/2	1 15/32	1 15/32	1 13/32	1 11/32	1 1/4
*1 1/4 to 1 1/2	0.750	4 1/4	4 3/8	4 1/2	4 1/2	4 5/8	4 5/8	1 13/16	1 13/16	1 25/32	1 3/4	1 11/16	1 5/8	1 1/4
*1 1/2	0.875	4 3/8	4 1/2	4 5/8	4 5/8	4 3/4	4 3/4	1 13/16	1 13/16	1 25/32	1 3/4	1 11/16	1 5/8	1 1/4
1 1/2 to 1 3/4	0.875	4 1/2	4 5/8	4 3/4	4 7/8	5	5	2 3/32	2 3/32	2 1/16	2 1/32	1 31/32	1 7/8	1 1/4
*1 3/4	1.000	5	5 1/8	5 1/4	5 3/8	5 1/2	5 1/2	2 3/32	2 3/32	2 1/16	2 1/32	1 31/32	1 7/8	1 1/2
1 3/4 to 2	1.000	5 1/8	5 1/4	5 3/8	5 1/2	5 5/8	5 5/8	2 13/32	2 13/32	2 3/8	2 5/16	2 1/4	2 5/32	1 1/2
*2	1.125	5 3/8	5 1/2	5 5/8	5 3/4	5 7/8	5 7/8	2 13/32	2 13/32	2 3/8	2 5/16	2 1/4	2 5/32	1 1/2
2 to 2 1/2	1.125	5 1/2	5 3/4	5 7/8	6	6 1/8	6 1/4	3	3	2 15/16	2 29/32	2 3/4	2 11/16	1 1/2
*2 1/2	1.563	6 3/8	6 5/8	6 3/4	6 7/8	7	7 1/8	3	3	2 15/16	2 29/32	2 3/4	2 11/16	1 3/4
2 1/2 to 3	1.563	6 3/4	7	7 1/8	7 1/4	7 1/4	7 1/2	3 19/32	3 19/32	3 17/32	3 15/32	3 11/32	3 7/32	1 3/4
*3	1.900	7 1/2	7 3/4	7 7/8	8	8	8 1/4	3 19/32	3 19/32	3 17/32	3 15/32	3 11/32	3 7/32	2

Cutters marked * will fill all requirements for chains as made in the United States at the present time.

$$F = D\left[0.8 \cos\left(18 \text{ deg.} - \frac{56}{T}\right) + 1.24 \cos\left(17 \text{ deg.} + \frac{116}{T} - Yab\right) - 1.3025\right] - 0.0015 \text{ in.}$$

$$cb = c'b' = \sqrt{[(E+F)^2 + (yz)^2]}$$

Width of cutter $= 1.02\,P\ [0.6 + \cot\ (180 \text{ deg.}/n)] \sin\ (180 \text{ deg.}/n)$ to the next higher 1/32 in.

Where the same roller diameter is commonly used with chains of two different pitches it is recommended that stock cutters be made wide enough to cut sprockets for both chains.

Marking of Cutters.—All cutters are to be marked, giving pitch, roller diameter and range of teeth to be cut.

Bores for Sprocket Cutters (Recommended Practice) are approximately as calculated from the formula

Bore $= 0.7 \sqrt{(\text{Width of Cutter} + \text{Roller Diameter} + 0.7\,P)}$, where P is the pitch.

Minimum Outside Diameters of Space Cutters for 35 teeth and over (Recommended Practice) are approximately as calculated from the formula

Outside Diameter $= 1.2\ (\text{Bore} + \text{Roller Diameter} + 0.7\,P) + 1$ in.

For less than 35 teeth the diameters can sometimes be smaller as given in Table 9.

Design for Standard "Straddle Cutters" for Sprockets

Only two cutters of this type are required to cover the entire range of teeth. Cutter B is based on 11 teeth and is designed to be used for 17 teeth and less. The maximum pressure angle for new chain is 24.1 deg. and the average pressure angle is 17.6 deg. Cutter A is based on 40 teeth and is designed to be used for 18 teeth and over. The maximum pressure angle is 32 deg. and the average pressure angle is 23.7 deg. In Fig. 7 the pressure angle for a new chain would be the angle bax, and in Fig. 8 it would be $b'ax$.

P = pitch, D = roller diameter and T = number of teeth on which cutter is based.

Construction.—Draw XY and the two seating-curve circles as shown in Fig. 10, making radius $ax = 0.5025D + 0.0015$ in.

Locate c and c' from the dimensions K and J as given in Table 10. Locate b and b'. Draw cax and $c'a'x'$ and with centers c and c' draw the working curves xy and $x'y'$. Draw yz and $y'z'$ perpendicular to cy and $c'y'$ respectively. Draw

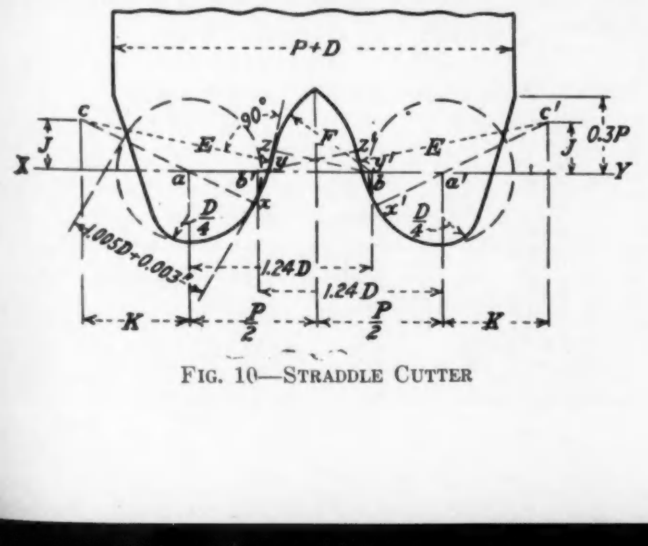


FIG. 10—STRADDLE CUTTER

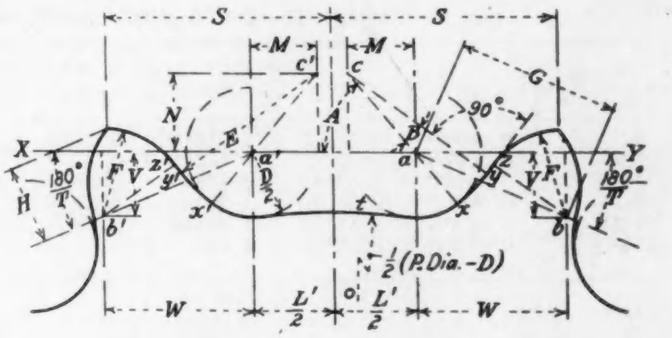


FIG. 11—TOOTH FORM FOR BLOCK CHAINS

bz and $b'z'$ parallel to cy and $c'y'$ respectively. With b and b' as centers and radius equal to bz, strike the arcs of the topping curves. The dimensions given in Table 10 may be calculated from the formulas given in connection with Fig. 8.

The maximum pressure angle $xab = 35 \text{ deg.} - (120/T)$

The minimum pressure angle $xab - acy = 17 \text{ deg.} - (64/T)$

The average pressure angle $= 26 \text{ deg.} - (92/T)$

Cutter width is $P + D$. Cutter bores and keyways are as given in Table 9 and cutter diameters are as given in Table 9 for 35 teeth and more.

TABLE 10—DATA FOR STRADDLE-CUTTER OUTLINES

Cutter	To Cut No. of Teeth T	K	J	F	Chord xy	yz	E
B	17 and under	$0.730D$	$0.327D$	$0.6937D - 0.0015$	$0.2928D + 0.0003$	$0.0617D$	$1.3025D + 0.0015$
A	18 and over	$0.678D$	$0.424D$	$0.6596D - 0.0015$	$0.3762D + 0.0004$	$0.1007D$	$1.3025D + 0.0015$

Standard Sprocket-Tooth Form for Block Chains or Twin-Roller Chains

The tooth form for block and twin-roller chains is designed to follow the essential specifications as for standard roller chains with the same angle of bend.

L = pitch of blocks; e = pitch of side plates; P, nominal pitch of chain $= L + e$; D = diameter of round end of block; T = number of teeth.

$$L' = L + 0.005D + 0.003 \text{ in.}$$
$$A = 35 \text{ deg.} + (120/T)$$
$$B = 18 \text{ deg.} - (28/T)$$
$$ac = 0.8D$$
$$M = 0.8D \cos\ [35 \text{ deg.} + (120/T)]$$
$$N = 0.8D \sin\ [35 \text{ deg.} + (120/T)]$$
$$E = 1.3D$$
$$\text{Chord } xy = 2.6D \sin\ [9 \text{ deg.} - (14/T)]$$
$$yz = D\left[1.24 \sin\left(17 \text{ deg.} - \frac{32}{T}\right) - 0.8 \sin\left(18 \text{ deg.} - \frac{28}{T}\right)\right]$$
$$G = 1.24D$$
$$W = 1.24D \cos\ (180 \text{ deg.}/T)$$
$$V = 1.24D \sin\ (180 \text{ deg.}/T)$$
$$F = D\left[0.8 \cos\left(18 \text{ deg.} - \frac{28}{T}\right) + 1.24 \cos\left(17 \text{ deg.} - \frac{32}{T}\right) - 1.3\right]$$
$$H = \sqrt{F^2 - \left(1.24D - \frac{P - L'}{2}\right)^2} \text{ very nearly}$$
$$S = H \sin\left(\frac{180 \text{ deg.}}{T}\right) + \left(\frac{P - L'}{2}\right) \cos\left(\frac{180 \text{ deg.}}{T}\right) + \frac{L'}{2}, \text{ very nearly}$$

TABLE 11—DATA FOR LAYING OUT BLOCK-CUTTER OUTLINES

Range of Teeth	M	N	W	V	F	Chord xy	yz	ot	Q
6	$0.4589D$	$0.6554D$	$1.0738D$	$0.620D$	$0.6927D$	$0.3019D$	$0.0662D$	$0.97P - 0.5D$	$1.12P$
7–8	$0.5027D$	$0.6223D$	$1.1327D$	$0.504D$	$0.6845D$	$0.3227D$	$0.0779D$	$1.28P - 0.5D$	$1.12P$
9–11	$0.5348D$	$0.5950D$	$1.1782D$	$0.387D$	$0.6764D$	$0.3432D$	$0.0861D$	$1.76P - 0.5D$	$1.11P$
12–17	$0.5801D$	$0.5510D$	$1.2128D$	$0.258D$	$0.6638D$	$0.3622D$	$0.1111D$	$2.71P - 0.5D$	$1.10P$
18–34	$0.6120D$	$0.5152D$	$1.2353D$	$0.108D$	$0.6431D$	$0.3801D$	$0.1577D$	$5.41P - 0.5D$	$1.08P$
35 and up	$0.6403D$	$0.4796D$	$1.24D$	0	$0.6308D$	$0.3955D$	$0.1763D$	0	$1.05P$

Outside diameter of sprocket when tooth is pointed

$$\text{Outside diameter} = 2H + \frac{L' + (P - L')\cos(180\text{ deg.}/T)}{\sin(180\text{ deg.}/T)}$$

Minimum outside diameter of blank =

$$0.6e + \frac{L + e\cos(180\text{ deg.}/T)}{\sin(180\text{ deg.}/T)}$$

The pressure angle for a new chain $xab = 35\text{ deg.} - (60/T)$

The minimum pressure angle $xab - B = 17\text{ deg.} - (32/T)$

The average pressure angle $= 26\text{ deg.} - (46/T)$

Design for Standard Cutter for Block-Chain Sprockets

Sprocket cutters for block and twin-roller chains are made for the same ranges of teeth as for roller-chain sprockets and are based upon the same number of teeth.

Construction.—Referring to Fig. 12, draw XY. Locate a and a', making L' = pitch of block + 0.005 D + 0.003 in., and with these points as centers, with radius $D/2$ draw the circular arcs for the seating curves. With radius ot equal to one-half the bottom diameter of the largest sprocket to be cut, draw an arc tangent to the two seating curves. Locate c and c' from dimensions M and N in Table 11. With c and c' as centers describe the arcs xy and $x'y'$. Draw yz and $y'z'$ perpendicular to cy and $c'y'$ respectively. Locate b and b' from dimensions W and V in Table 11. With these as centers and a radius F, draw the arcs ZS and $Z'S'$. The line yz is a common tangent to the curves xy and zs.

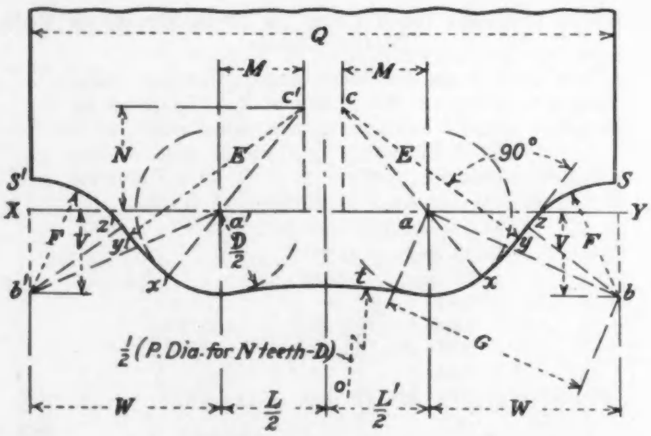


Fig. 12—Block Cutter

TABLE 12—CHECKS

Range of Teeth	E	G	Angle Yab
6	$1.3D$	$1.24D$	30 deg.
7–8	$1.3D$	$1.24D$	24 deg.
9–11	$1.3D$	$1.24D$	18 deg. 10 min.
12–17	$1.3D$	$1.24D$	12 deg.
18–34	$1.3D$	$1.24D$	5 deg.
35 up	$1.3D$	$1.24D$	0 deg.

Supplementary Data. Formulas used in the calculations of data in Tables 11 and 12 are given below.

The angle Yab is 180 deg./T when the cutter is made for T teeth only, but it has values as given in Table 12 for cutters covering a range of teeth as here designed. Accordingly the following formulas are special for cutters covering the standard ranges of teeth;

$$V = 1.24D \sin Yab$$

$$W = 1.24\,D \cos Yab$$

$$yx = D\left[1.24 \sin\left(17\text{ deg.} + \frac{148}{T} - Yab\right) - 0.8 \sin\left(18\text{ deg.} - \frac{28}{T}\right)\right]$$

$$F = D\left[0.8 \cos\left(18\text{ deg.} - \frac{28}{T}\right) + 1.24 \cos\left(17\text{ deg.} + \frac{148}{T} - Yab\right) - 1.3\right]$$

$$cb = c'b' = \sqrt{[(E+F)^2 + (yz)^2]}$$

The width of cutter $Q = 1.02 \sin (180\text{ deg.}/n) \times$ the outside diameter of the smallest sprocket in the range to be cut, to the next higher 1/32 in.

SUPPLEMENTARY INFORMATION

The following information relating to good roller-chain practice is more or less flexible and governed by the conditions obtaining in individual cases of application and in chain manufacture and therefore is not to be considered as American Standard.

TABLE 13—RECOMMENDED SIZES OF CUTTERS FOR BLOCK-CHAIN SPROCKETS

Pitch of Chain	Block End Diameter	Pitch of Block	Cutter Diameter	Bore	Keyway	Cutter Width, Number of Teeth T: 6	7–8	9–11	12–17	18–34	35 and more
1	0.325	0.400	3	1	5/32 x 5/64	1 3/16	1 5/32	1 5/32	1 1/8	1 1/8	1 3/32
1 1/2	0.531	0.563	4	1 1/4	3/16 x 3/32	1 25/32	1 3/4	1 3/4	1 11/16	1 11/16	1 5/8

Working Loads and Horsepowers

These tables are based on sprockets having not less than 15 teeth.

TABLE 14—STANDARD ROLLER CHAIN SERIES

Chain No.	Pitch, In.	Maximum R.P.M.	The Upper Figures are Rated Horsepowers The Lower Figures are Loads in Pounds Chain Velocity in Feet per Minute									
			50	100	200	300	400	600	800	1,000	1,300	1,600
35 *N*	3/8	3,620	0.25 162	0.45 150	0.79 132	1.06 117	1.27 105	1.60 88	1.82 75	1.94 64	2.07 52	2.10 43
40	1/2	2,717	0.41 269	0.76 250	1.33 219	1.78 194	2.12 175	2.65 146	2.98 123	3.25 107	3.41 87	3.44 71
50	5/8	1,927	0.64 428	1.20 397	2.11 348	2.82 308	3.38 278	4.22 232	4.75 196	5.13 169	5.4 137	5.47 113
60	3/4	1,454	0.93 645	1.80 597	3.03 524	4.06 462	4.86 419	6.07 348	7.15 295	7.68 253	8.10 206	8.20 169
80	1	941	1.66 1,092	3.07 1,014	5.58 891	7.20 785	8.64 712	10.8 592	12.1 500	13.0 431	13.75 349	13.85 286
100	1 1/4	652	2.41 1,592	4.48 1,478	7.88 1,294	10.5 1,153	12.6 1,035	15.7 862	17.6 727	18.85 622	19.9 506	20.0 412
120	1 1/2	515	3.63 2,415	6.76 2,230	11.8 1,960	15.8 1,730	18.9 1,565	23.6 1,306	26.5 1,098	28.7 946	30.2 766	30.2 623
140	1 3/4	375	4.36 2,880	8.11 2,674	14.2 2,330	19.0 2,070	22.8 1,870	28.4 1,560	31.7 1,308	34.0 1,127	36.0 914	36.0 742
160	2	316	5.96 3,940	11.10 3,658	19.4 3,202	25.8 2,840	31.1 2,560	38.8 2,130	43.4 1,792	46.7 1,540	49.0 1,244	49.0 1,013
200	2 1/2	224	10.06 6,636	18.70 6,161	32.7 5,390	43.6 4,790	52.3 4,320	65.4 3,600	73.5 3,030	79.0 2,605	84.0 2,130	84.0 1,742

The chain manufacturer should be consulted (*a*) if there are less than 15 teeth on the smaller sprocket, (*b*) if the speed of the smaller sprocket exceeds 80 per cent of the maximum number of revolutions per minute given or (*c*) if the chain speeds exceeds 1000 ft. per min.

TABLE 15—EXTRA-HEAVY ROLLER CHAIN SERIES

Chain No.	Pitch, In.	Maximum R.P.M.	The Upper Figures are Rated Horsepowers The Lower Figures are Loads in Pounds Chain Velocity in Feet per Minute									
			50	100	200	300	400	600	800	1,000	1,300	1,600
60 *H*	3/4	1,306	1.07 702	1.97 652	3.46 570	4.62 507	5.53 457	6.92 382	7.76 321	8.35 276	8.84 224	8.84 182
80 *H*	1	871	1.77 1,170	3.29 1,085	5.76 950	7.68 840	9.20 760	11.5 634	12.9 533	13.9 459	14.7 372	14.70 303
100 *H*	1 1/4	595	2 55 1,680	4.73 1,565	8.30 1,368	11.1 1,215	13.3 1,093	16.6 914	18.6 767	20.0 659	21.0 534	21.0 433
120 *H*	1 1/2	471	3.79 2,500	7.03 2,320	12.4 2,032	16.4 1,810	19.7 1,630	24.6 1,360	27.7 1,144	29.7 982	31.3 796	31.3 644
140 *H*	1 3/4	350	4.55 3,000	8.45 2,780	14.8 2,440	19.7 2,170	23.7 1,950	29.6 1,630	33.2 1,366	35.6 1,175	37.5 952	37.5 771
160 *H*	2	297	6.18 4,080	11.5 3,790	20.1 3,320	26.8 2,880	32.2 2,650	40.2 2,210	44.8 1,853	48.3 1,596	50.6 1,287	50.6 1,042
200 *H*	2 1/2	213	10.7 7,020	19.7 6,520	34.6 5,700	46.2 5,070	55.3 4,560	69.2 3,810	77.5 3,199	83.5 2,754	88.5 2,248	88.5 1,828

The chain manufacturer should be consulted (*a*) if there are less than 15 teeth on the smaller sprocket, (*b*) if the speed of the smaller sprocket exceeds 80 per cent of the maximum number of revolutions per minute given or (*c*) if the chain speed exceeds 1000 ft. per min.

Recommended Working Loads

As a guide in the selection of chains, the following formulas are given for the calculation of permissible working loads.

$$T = [2{,}600{,}000\,A/(V + 600)] - [WV^2/115{,}900] \qquad (16)$$
$$\text{Hp.} = T\,V/33{,}000$$

T being the working load in pounds; A the projected pin-bearing area, or diameter of pin times length of bushing; V the chain velocity in feet per minute and W the weight in pounds per foot of chain.

The second fraction in formula (16) represents the centrifugal pull Wv^2/g which is deducted when the chain velocity is 800 ft. per min. or more.

Under exceptional conditions these values may be increased by as much as 25 per cent, but only when the lubrication and installation as well as the design of the drive is correct. On the other hand, where the conditions of lubrication are poor, where the chain is subject to suddenly applied loads, where the service is continuous, where unusually long life is required or where the sprockets are subject to misalignment due to weaving, the chains should be selected on the basis of approximately half of the above ratings.

Sprocket Speeds

The maximum allowable impact between the chain rollers and the sprocket teeth limits the sprocket speed to

$$\text{R.P.M.} = (1920/P)\sqrt{(A/WP)}$$

P being the pitch; A the projected area of the roller, or diameter times length and W the weight per foot of chain.

The maximum revolutions per minute of sprockets as calculated from this formula for standard roller chains are given in the third column of Table 14 and for the extra heavy series they are given in Table 15. Sprocket speeds as high as those calculated from the above formula may not be permissible if the resulting chain velocity is not too high.

Metal Patterns for Cast Sprockets

Bottom Diameter of Pattern.—After allowing for shrinkage on bottom diameter subtract 1/32 inch to allow for roughness of teeth and rapping.

Pitch-Line Clearance.—After cutting all of the teeth to the proper depth with a standard sprocket-cutter, rotate the blank on the work arbor an amount equal to $0.02\,P$ + 0.02 in. at the pitch line. Then take a second cut all around.

Standard Pitch-Diameters and Outside Diameters of Sprockets

TABLE 16—SPROCKET DIAMETERS FOR 1-IN. PITCH CHAINS

No. of Teeth	Pitch Diameter	Outside Diameter	No. of Teeth	Pitch Diameter	Outside Diameter	No. of Teeth	Pitch Diameter	Outside Diameter
6	2.0000	2.332	46	14.6536	15.219	86	27.3807	27.962
7	2.3048	2.677	47	14.9717	15.538	87	27.6989	28.281
8	2.6131	3.014	48	15.2898	15.857	88	28.0171	28.599
9	2.9238	3.347	49	15.6079	16.176	89	28.3354	28.918
10	3.2361	3.678	50	15.9260	16.495	90	28.6536	29.236
11	3.5495	4.006	51	16.2441	16.813	91	28.9718	29.554
12	3.8637	4.332	52	16.5619	17.132	92	29.2900	29.873
13	4.1785	4.657	53	16.8803	17.451	93	29.6082	30.191
14	4.4940	4.982	54	17.1984	17.769	94	29.9264	30.510
15	4.8097	5.305	55	17.5166	18.088	95	30.2446	30.828
16	5.1259	5.627	56	17.8347	18.406	96	30.5628	31.146
17	5.4423	5.950	57	18.1529	18.725	97	30.8811	31.465
18	5.7588	6.271	58	18.4710	19.044	98	31.1994	31.783
19	6.0756	6.593	59	18.7892	19.363	99	31.5177	32.102
20	6.3925	6.914	60	19.1073	19.681	100	31.8360	32.420
21	6.7095	7.235	61	19.4255	20.000	101	32.1543	32.739
22	7.0266	7.555	62	19.7437	20.318	102	32.4726	33.057
23	7.3439	7.875	63	20.0618	20.637	103	32.7909	33.376
24	7.6613	8.196	64	20.3800	20.955	104	33.1091	33.694
25	7.9787	8.516	65	20.6982	21.274	105	33.4274	34.012
26	8.2962	8.836	66	21.0164	21.593	106	33.7457	34.331
27	8.6138	9.156	67	21.3346	21.911	107	34.0640	34.649
28	8.9315	9.475	68	21.6528	22.230	108	34.3823	34.968
29	9.2491	9.795	69	21.9710	22.548	109	34.7006	35.286
30	9.5668	10.114	70	22.2892	22.867	110	35.0189	35.605
31	9.8845	10.434	71	22.6074	23.185	111	35.3371	35.923
32	10.2023	10.753	72	22.9256	23.504	112	35.655	36.241
33	10.5201	11.072	73	23.2438	23.822	113	35.974	36.560
34	10.8380	11.392	74	23.5620	24.141	114	36.292	36.878
35	11.1558	11.711	75	23.8802	24.459	115	36.610	37.197
36	11.4737	12.030	76	24.1984	24.778	116	36.929	37.515
37	11.7917	12.349	77	24.5166	25.096	117	37.247	37.833
38	12.1096	12.668	78	24.8349	25.415	118	37.565	38.152
39	12.4275	12.987	79	25.1531	25.733	119	37.883	38.470
40	12.7455	13.306	80	25.4713	26.052	120	38.201	38.788
41	13.0635	13.625	81	25.7895	26.370	121	38.519	39.106
42	13.3815	13.944	82	26.1078	26.689	122	38.837	39.425
43	13.6995	14.263	83	26.4260	27.007	123	39.156	39.743
44	14.0175	14.582	84	26.7443	27.326	124	39.475	40.062
45	14.3356	14.901	85	27.0625	27.644	125	39.794	40.381

For other pitches multiply these values by the pitch.
To obtain bottom diameters subtract diameter of chain roller from pitch diameter.

TABLE 17—PITCH-LINE CLEARANCES FOR METAL PATTERNS CUT WITH STANDARD SPROCKET-CUTTERS

Pitch, In.	½	⅝	¾	1	1¼	1½	1¾	2	2½
Clearance, In.	0.030	0.033	0.035	0.040	0.045	0.050	0.055	0.060	0.070

Pattern Teeth for Cast Sprockets Made from Wooden Patterns

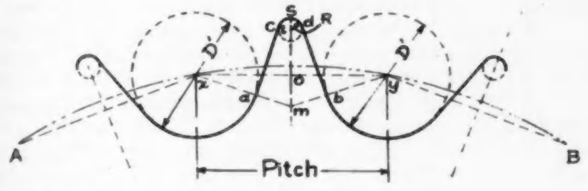


FIG. 13

TABLE 18—DIMENSIONS OF SPROCKET TEETH FOR WOODEN PATTERNS

Pitch, In.	Roller Diameter	*os*	*D'*	*ot*	*R*
½	0.313	0.150	11/32	⅛	0.02
⅝	0.400	0.187	7/16	11/64	0.02
¾	0.469	0.225	½	3/16	0.04
1	0.625	0.300	21/32	15/64	0.05
1¼	0.750	0.375	25/32	17/64	0.10
1½	0.875	0.450	59/64	19/64	9/64
1¾	1.000	0.525	1 3/64	21/64	3/16
2	1.125	0.600	1 11/64	23/64	15/64
2½	1.563	0.750	1⅝	9/16	11/64
3	1.900	0.900	1 31/32	11/16	3/16

AB (Fig. 13) is the pitch circle drawn with shrinkage allowance.

$D' = 1.02\,D + 0.02$ in. where D is the roller diameter and xy the pitch of the chain.

Som is a perpendicular bisector of xy.

$R = 0.56\,P - 0.76\,D'$

$os = 0.3\,P$

ac and *bd* are straight lines tangent to the circular arcs.

Bottom diameter of pattern = Pitch Diameter + Shrinkage $- 0.02\,D - 0.02$ in.

Aircraft-Engine Division

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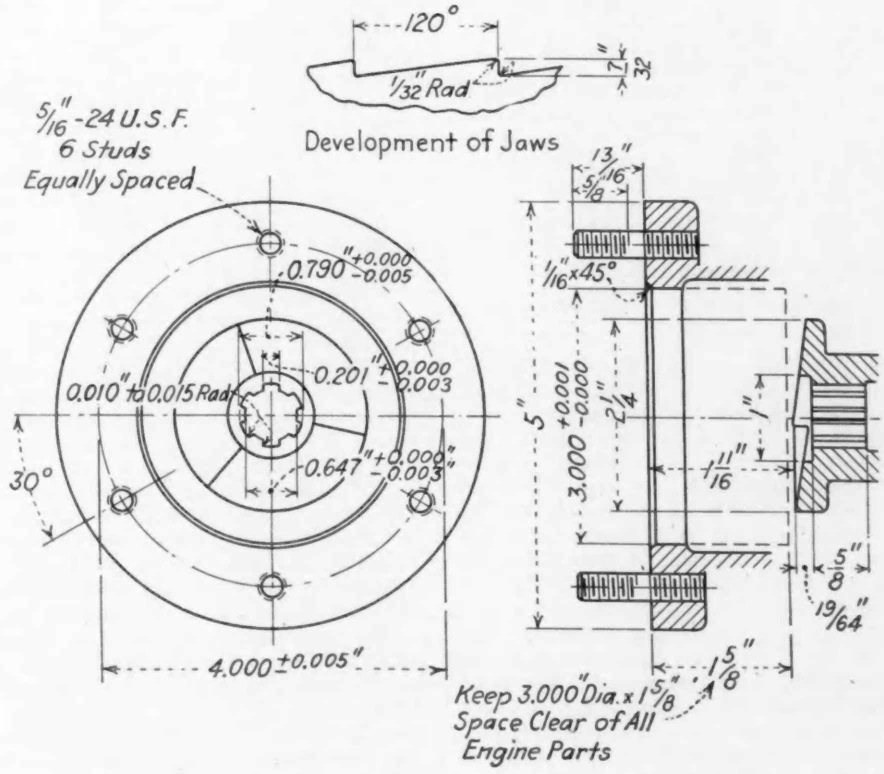
Aircraft-Starter Mounting

(*Proposed S.A.E. Standard*)

This small flange as illustrated was submitted to aircraft-engine manufacturers and to the Aircraft-Engine Division for consideration and letter-ballot as to whether the Standards Committee should be asked to approve this as an addition to the present large-size starter-mounting standard. The Division form records one dissenting vote and one member not voting. The survey of engine manufacturers shows 24 engine manufacturers in favor of the adoption of this specification and 2 opposed.

As a result the Standards Committee is requested to approve the accompanying specification as an additional S.A.E. Standard on Aircraft-Starter Mountings.



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Edward P. Warner, *Chairman*	McGraw-Hill Publishing Co., Inc.
Mac Short, *Vice-Chairman*	Stearman Aircraft Co.
Don M. Alexander	Alexander Aircraft Co.
Lieut. R. S. Barnaby	Bureau of Aeronautics, Navy Department
John R. Cautley	Bendix Brake Co.
G. G. Emerson	Wright Aeronautical Corp.
J. F. Hardecker	Naval Aircraft Factory
Lieut. C. B. Harper	Bureau of Aeronautics, Navy Department
H. A. Hicks	Stout Metal Airplane Co.
Major C. W. Howard	Air Corps
I. M. Laddon	Consolidated Aircraft Corp.
K. M. Lane	Department of Commerce
B. J. Lemon	United States Rubber Co.
R. G. Lockwood	Fairchild Airplane Mfg. Corp.
C. J. McCarthy	Chance Vought Corp.
C. N. Montieth	Boeing Airplane Co.
C. T. Porter	Keystone Aircraft Corp.
L. D. Seymour	National Air Transport, Inc.
J. W. Swain	Firestone Steel Products Co.
Edward Wallace	Great Lakes Aircraft Corp.
T. P. Wright	Curtiss Aeroplane & Motor Co., Inc.

Airplane-Wheel Rims

(*Proposed Revision of S.A.E. Recommended Practice*)

To bring the S.A.E. Recommended Practice on Airplane-Wheel Rims in accord with the Tire and Rim Association specifications, the Division recommends the change in the ledge widths on the No. 10 and No. 12 sizes from 1-3/16 to 1-7/32 in. and 1-3/8 to 1-29/64 in. respectively.

Ball and Roller Bearings Division

PERSONNEL

H. E. Brunner, *Chairman*	S. K. F. Industries, Inc.
G. R. Bott, *Vice-Chairman*	Norma-Hoffmann Bearings Corp.
F. H. Buhlmann	Rollway Bearing Co., Inc.
E. R. Carter, Jr.	Fafnir Bearing Co.
D. F. Chambers	Bearing Co of America.
L. A. Cummings	Marlin-Rockwell Corp.
T. C. Delaval-Crow	New Departure Mfg. Co.
H. R. Gibbons	Hyatt Bearings Division, General Motors Corp.
B. H. Gilpin	Pratt &. Whitney Aircraft Co.
G. E. Parker	Cadillac Motor Car Co.
H. N. Parsons	Strom Bearings Co.
Ernest Wooler	Timken Roller Bearing Co.

Annular Ball Bearings—Single-Row Type

(*Proposed American Standard and Proposed Revision of S.A.E. Standard*)

When the Sectional Committee on Ball Bearings was organized in November, 1919, with the Society and the American Society of Mechanical Engineers as sponsors under the procedure of the American Engineering Standards Committee, now the American Standards Association, the principal purpose was to cooperate with the national standardizing bodies in foreign countries in developing, so far as possible, ball-bearing standards that would be in international agreement. The standards for the light, medium and heavy series of single-row annular ball-bearings that had been established by the Society for the automotive industries were taken as the basis of the Sectional Committee's work and a sub-committee was appointed to secure further information regarding American and foreign practice. It was found that several differences existed in these, some of which were major in character. Negotiations continued for several years which resulted in the tentative approval of a complete list of this type of bearing in sizes in the three series that extend considerably beyond the sizes commonly used for automotive applications. It is felt that the proposed tables include all sizes of bearing in these series that are in common usage and the Sectional Committee has approved the report for submission to the sponsors and the American Standards Association for final approval and adoption as American Standard.

The report has been carefully considered by the Ball and Roller Bearings Division of the Standards Committee, the major part of whose personnel is included also on the Sectional Committee on Ball Bearings, and the Division now recommends that the report be approved by the Society as a sponsor. As the report of the Sectional Committee was being formulated in the last few years, a number of changes were made that varied from the S.A.E. Standard tables, but the members of the Ball and Roller Bearings Division felt that the S.A.E. Standard should not be revised until the Sectional Committee's report was completed. This has caused some confusion and the Division now recommends that the S.A.E. Standard for Annular Ball Bearings of the Single-Row Type in the light, medium and heavy series, printed on pp. 254, 255, 256 and 259 of the 1929 edition of the S.A.E. HANDBOOK be revised in accordance with the Sectional Committee's report, maintaining in the S.A.E. Standard only those ranges of sizes now given therein.

Annular Ball and Roller Bearings—Wide Type

(*Proposed American Recommended Practice*)

In June, 1927, a meeting of anti-friction bearing and industrial electric-motor manufacturers was held by the National Electrical Manufacturers Association to discuss the standardization of anti-friction bearings for electric motors that resulted in referring the project to the Ball and Roller Bearings Division of the Society for recommendations. It was suggested at the meeting that the project be referred to the Sectional Committee on Ball Bearings, but it was felt that the Ball and Roller Bearings Division, being more experienced in this work, should handle it. The project was approved by the Council of the Society as a special one of the Division and two sub-committees were organized, one to study the roller bearings and the other the ball bearings. Each sub-committee, after securing sufficient data, drafted its preliminary recommendation and referred it to the other sub-committee. The two sub-committees then consolidated their recommendations into a single report that was submitted to the Ball and Roller Bearings Division which gave it wide circulation for comment. The National Electrical Manufacturers Association, to which the report was also referred, has reported approval of the report by its motor and generator section and by its general standards committee at their meetings last June.

On completing the report, the Ball and Roller Bearings Division felt that in submitting it for final approval as American Recommended Practice it should come from the Sectional Committee on Ball Bearings rather than the Division inasmuch as the recommendation was intended primarily for industrial electric-motors. Accordingly the report was referred to the Sectional Committee and approved by it for submission to the sponsors and the American Standards Association for final approval. In accordance with regular S.A.E. Standards procedure the report is therefore submitted by the Ball and Roller Bearings Division for its approval by the Society as sponsors, for submission to the American Standards Association and final adoption as American Recommended Practice.

The report is in two sections; the regular series and the extended series of sizes, the regular series being the same as the wide type annular bearings printed on p. 257 of the 1929 edition of the S.A.E. HANDBOOK. The extended series was developed for heavier installations. The S.A.E. table has been expanded to include all detail dimensions of the bearings required for their interchangeability in order to

ANNULAR BALL BEARINGS
SINGLE ROW TYPE

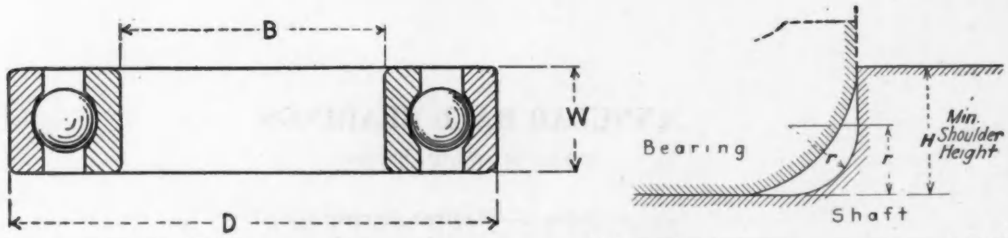


Table 1—Light Series

Bearing Number	Bore B			Outside Diameter D			Width of Individual Rings W			r Shaft and Housing Fillet Radii[1] Maximum		H Height of Shoulder on Shaft Minimum		Eccentricity Tolerances, Inches Maximum	
	Nominal Diameter		Toler., In. +0.0000	Nominal Diameter		Toler., In. +0.0000	Nominal Width		Toler., In. +0.000						
	Mm.	In.		Mm.	In.		Mm.	In.		Mm.	In.	Mm.	In.	Inner Ring	Outer Ring
34	4	0.1575	−0.0004	16	0.6299	−0.0004	5	0.1969	−0.005	0.4	0.016	1.0	0.039	0.0004	0.0008
35	5	0.1969	0.0004	19	0.7480	0.0004	6	0.2362	0.005	0.4	0.016	1.0	0.039	0.0004	0.0008
37	7	0.2756	0.0004	22	0.8661	0.0004	7	0.2756	0.005	0.4	0.016	1.0	0.039	0.0004	0.0008
39	9	0.3543	0.0004	26	1.0236	0.0004	8	0.3150	0.005	0.4	0.016	1.0	0.039	0.0004	0.0008
200	10	0.3937	0.0004	30	1.1811	0.0004	9	0.3543	0.005	0.6	0.024	2.5	0.098	0.0006	0.0012
201	12	0.4724	0.0004	32	1.2598	0.0005	10	0.3937	0.005	0.6	0.024	2.5	0.098	0.0006	0.0012
202	15	0.5906	0.0004	35	1.3780	0.0005	11	0.4331	0.005	0.6	0.024	2.5	0.098	0.0006	0.0012
203	17	0.6693	0.0004	40	1.5748	0.0005	12	0.4724	0.005	1.0	0.039	3.0	0.118	0.0006	0.0012
204	20	0.7874	0.0004	47	1.8504	0.0005	14	0.5512	0.005	1.0	0.039	3.0	0.118	0.0006	0.0012
205	25	0.9843	0.0004	52	2.0472	0.0006	15	0.5906	0.005	1.0	0.039	3.0	0.118	0.0008	0.0012
206	30	1.1811	0.0004	62	2.4409	0.0006	16	0.6299	0.005	1.0	0.039	3.0	0.118	0.0008	0.0012
207	35	1.3780	0.0005	72	2.8346	0.0006	17	0.6693	0.005	1.0	0.039	3.5	0.138	0.0008	0.0012
208	40	1.5748	0.0005	80	3.1496	0.0006	18	0.7087	0.005	1.0	0.039	3.5	0.138	0.0008	0.0012
209	45	1.7717	0.0005	85	3.3465	0.0008	19	0.7480	0.005	1.0	0.039	3.5	0.138	0.0010	0.0016
210	50	1.9685	0.0005	90	3.5433	0.0008	20	0.7874	0.005	1.0	0.039	3.5	0.138	0.0010	0.0016
211	55	2.1654	0.0006	100	3.9370	0.0008	21	0.8268	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
212	60	2.3622	0.0006	110	4.3307	0.0008	22	0.8661	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
213	65	2.5591	0.0006	120	4.7244	0.0008	23	0.9055	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
214	70	2.7559	0.0006	125	4.9213	0.0010	24	0.9449	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
215	75	2.9528	0.0006	130	5.1181	0.0010	25	0.9843	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
216	80	3.1496	0.0006	140	5.5118	0.0010	26	1.0236	0.005	2.0	0.079	5.0	0.197	0.0012	0.0018
217	85	3.3465	0.0008	150	5.9055	0.0010	28	1.1024	0.005	2.0	0.079	5.0	0.197	0.0012	0.0018
218	90	3.5433	0.0008	160	6.2992	0.0010	30	1.1811	0.005	2.0	0.079	5.0	0.197	0.0012	0.0018
219	95	3.7402	0.0008	170	6.6929	0.0010	32	1.2598	0.005	2.0	0.079	6.0	0.236	0.0012	0.0018
220	100	3.9370	0.0008	180	7.0866	0.0010	34	1.3386	0.005	2.0	0.079	6.0	0.236	0.0012	0.0018
221	105	4.1339	0.0008	190	7.4803	0.0012	36	1.4173	0.005	2.0	0.079	6.0	0.236	0.0012	0.0018
222	110	4.3307	0.0008	200	7.8740	0.0012	38	1.4961	0.005	2.0	0.079	6.0	0.236	0.0012	0.0018
224	120	4.7244	0.0008	215	8.4646	0.0012	40	1.5748	0.005	2.0	0.079	6.0	0.236	0.0014	0.0020
226	130	5.1181	0.0010	230	9.0551	0.0012	40	1.5748	0.005	2.5	0.098	7.0	0.276	0.0014	0.0020
228	140	5.5118	0.0010	250	9.8425	0.0012	42	1.6535	0.005	2.5	0.098	7.0	0.276	0.0014	0.0020
230	150	5.9055	0.0010	270	10.6299	0.0016	45	1.7717	0.005	2.5	0.098	7.0	0.276	0.0014	0.0020
232	160	6.2992	0.0010	290	11.4173	0.0016	48	1.8898	0.005	2.5	0.098	7.0	0.276	0.0014	0.0020
234	170	6.6929	0.0010	310	12.2047	0.0016	52	2.0472	0.005	3.0	0.118	9.0	0.354	0.0014	0.0020
236	180	7.0866	0.0010	320	12.5984	0.0016	52	2.0472	0.005	3.0	0.118	9.0	0.354	0.0014	0.0020
238	190	7.4803	0.0012	340	13.3858	0.0016	55	2.1654	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
240	200	7.8740	0.0012	360	14.1732	0.0016	58	2.2835	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
244	220	8.6614	0.0012	400	15.7480	0.0024	65	2.5591	0.010	3.0	0.118	9.0	0.354	0.0018	0.0024
248	240	9.4488	0.0012	440	17.3228	0.0024	72	2.8346	0.010	3.0	0.118	9.0	0.354	0.0018	0.0024
252	260	10.2362	0.0012	480	18.8976	0.0024	80	3.1496	0.010	4.0	0.157	11.0	0.433	0.0018	0.0024
256	280	11.0236	0.0016	500	19.6850	0.0024	80	3.1496	0.010	4.0	0.157	11.0	0.433	0.0018	0.0024
260	300	11.8110	0.0016	540	21.2598	0.0024	85	3.3465	0.010	4.0	0.157	11.0	0.433	0.0018	0.0024
264	320	12 5984	0.0016	580	22.8346	0.0024	92	3.6220	0.010	4.0	0.157	11.0	0.433	0.0018	0.0024

Note 1—The corner radius or chamfer on bearings must clear the maximum fillet radius given in the table and provide for sufficient bearing area against the minimum shoulder on the shafts.
Conversion from metric to decimal inch dimensions is according to the formula 1 mm. = 0.0393700 inch.

ANNULAR BALL BEARINGS

SINGLE ROW TYPE

Table 2—Medium Series

Bearing Number	Bore B			Outside Diameter D			Width of Individual Rings W			r Shaft and Housing Fillet Radii[1] Maximum		H Height of Shoulder on Shaft Minimum		Eccentricity Tolerances, Inches Maximum	
	Nominal Diameter		Toler., In. +0.0000	Nominal Diameter		Toler., In. +0.0000	Nominal Width		Toler., In. +0.000						
	Mm.	In.		Mm.	In.		Mm.	In.		Mm.	In.	Mm.	In.	Inner Ring	Outer Ring
300	10	0.3937	−0.0004	35	1.3780	−0.0005	11	0.4331	−0.005	0.6	0.024	2.5	0.098	0.0006	0.0012
301	12	0.4724	0.0004	37	1.4567	0.0005	12	0.4724	0.005	1.0	0.039	3.0	0.118	0.0006	0.0012
302	15	0.5906	0.0004	42	1.6535	0.0005	13	0.5118	0.005	1.0	0.039	3.0	0.118	0.0006	0.0012
303	17	0.6693	0.0004	47	1.8504	0.0005	14	0.5512	0.005	1.0	0.039	3.0	0.118	0.0006	0.0012
304	20	0.7874	0.0004	52	2.0472	0.0006	15	0.5906	0.005	1.0	0.039	3.5	0.138	0.0006	0.0012
305	25	0.9843	0.0004	62	2.4409	0.0006	17	0.6693	0.005	1.0	0.039	3.5	0.138	0.0008	0.0012
306	30	1.1811	0.0004	72	2.8346	0.0006	19	0.7480	0.005	1.0	0.039	3.5	0.138	0.0008	0.0012
307	35	1.3780	0.0005	80	3.1496	0.0006	21	0.8268	0.005	1.5	0.059	4.5	0.177	0.0008	0.0012
308	40	1.5748	0.0005	90	3.5433	0.0008	23	0.9055	0.005	1.5	0.059	4.5	0.177	0.0008	0.0012
309	45	1.7717	0.0005	100	3.9370	0.0008	25	0.9843	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
310	50	1.9685	0.0005	110	4.3307	0.0008	27	1.0630	0.005	2.0	0.079	5.0	0.197	0.0010	0.0016
311	55	2.1654	0.0006	120	4.7244	0.0008	29	1.1417	0.005	2.0	0.079	5.0	0.197	0.0010	0.0016
312	60	2.3622	0.0006	130	5.1181	0.0010	31	1.2205	0.005	2.0	0.079	6.0	0.236	0.0010	0.0016
313	65	2.5591	0.0006	140	5.5118	0.0010	33	1.2992	0.005	2.0	0.079	6.0	0.236	0.0010	0.0016
314	70	2.7559	0.0006	150	5.9055	0.0010	35	1.3780	0.005	2.0	0.079	6.0	0.236	0.0010	0.0016
315	75	2.9528	0.0006	160	6.2992	0.0010	37	1.4567	0.005	2.0	0.079	6.0	0.236	0.0010	0.0016
316	80	3.1496	0.0006	170	6.6929	0.0010	39	1.5354	0.005	2.0	0.079	6.0	0.236	0.0012	0.0018
317	85	3.3465	0.0008	180	7.0866	0.0010	41	1.6142	0.005	2.5	0.098	7.0	0.276	0.0012	0.0018
318	90	3.5433	0.0008	190	7.4803	0.0012	43	1.6929	0.005	2.5	0.098	7.0	0.276	0.0012	0.0018
319	95	3.7402	0.0008	200	7.8740	0.0012	45	1.7717	0.005	2.5	0.098	7.0	0.276	0.0012	0.0018
320	100	3.9370	0.0008	215	8.4646	0.0012	47	1.8504	0.005	2.5	0.098	7.0	0.276	0.0012	0.0018
321	105	4.1339	0.0008	225	8.8583	0.0012	49	1.9291	0.005	2.5	0.098	7.0	0.276	0.0012	0.0018
322	110	4.3307	0.0008	240	9.4488	0.0012	50	1.9685	0.005	2.5	0.098	7.0	0.276	0.0012	0.0018
324	120	4.7244	0.0008	260	10.2362	0.0012	55	2.1654	0.010	2.5	0.098	7.0	0.276	0.0014	0.0020
326	130	5.1181	0.0010	280	11.0236	0.0016	58	2.2835	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
328	140	5.5118	0.0010	300	11.8110	0.0016	62	2.4409	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
330	150	5.9055	0.0010	320	12.5984	0.0016	65	2.5591	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
332	160	6.2992	0.0010	340	13.3858	0.0016	68	2.6772	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
334	170	6.6929	0.0010	360	14.1732	0.0016	72	2.8346	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
336	180	7.0866	0.0010	380	14.9606	0.0024	75	2.9528	0.010	3.0	0.118	9.0	0.354	0.0014	0.0020
338	190	7.4803	0.0012	400	15.7480	0.0024	78	3.0709	0.010	4.0	0.157	11.0	0.433	0.0014	0.0020
340	200	7.8740	0.0012	420	16.5354	0.0024	80	3.1496	0.010	4.0	0.157	11.0	0.433	0.0014	0.0020
344	220	8.6614	0.0012	460	18.1102	0.0024	88	3.4646	0.010	4.0	0.157	11.0	0.433	0.0018	0.0024
348	240	9.4488	0.0012	500	19.6850	0.0024	95	3.7402	0.010	4.0	0.157	11.0	0.433	0.0018	0.0024
352	260	10.2362	0.0012	540	21.2598	0.0024	102	4.0157	0.010	5.0	0.197	14.0	0.551	0.0018	0.0024
356	280	11.0236	0.0016	580	22.8346	0.0024	108	4.2520	0.010	5.0	0.197	14.0	0.551	0.0018	0.0024

Note 1—The corner radius or chamfer on bearings must clear the maximum fillet radius given in the table and provide for sufficient bearing area against the minimum shoulder on the shafts.

Conversion from metric to decimal inch dimensions is according to the formula 1 mm. = 0.0393700 inch.

ANNULAR BALL BEARINGS

SINGLE ROW TYPE

Table 3—Heavy Series

Bearing Number	Bore B			Outside Diameter D			Width of Individual Rings W			r Shaft and Housing Fillet Radii[1] Maximum		H Height of Shoulder on Shaft Minimum		Eccentricity Tolerances, Inches Maximum	
	Nominal Diameter		Toler., In. +0.0000	Nominal Diameter		Toler., In. +0.0000	Nominal Width		Toler., In. +0.000						
	Mm.	In.		Mm.	In.		Mm.	In.		Mm.	In.	Mm.	In.	Inner Ring	Outer Ring
403	17	0.6693	−0.0004	62	2.4409	−0.0006	17	0.6693	−0.005	1.0	0.039	4.5	0.177	0.0006	0.0012
404	20	0.7874	0.0004	72	2.8346	0.0006	19	0.7480	0.005	1.0	0.039	4.5	0.177	0.0006	0.0012
405	25	0.9843	0.0004	80	3.1496	0.0006	21	0.8268	0.005	1.5	0.059	5.0	0.197	0.0008	0.0012
406	30	1.1811	0.0004	90	3.5433	0.0008	23	0.9055	0.005	1.5	0.059	5.0	0.197	0.0008	0.0012
407	35	1.3780	0.0005	100	3.9370	0.0008	25	0.9843	0.005	1.5	0.059	5.0	0.197	0.0008	0.0012
408	40	1.5748	0.0005	110	4.3307	0.0008	27	1.0630	0.005	2.0	0.079	5.5	0.217	0.0008	0.0012
409	45	1.7717	0.0005	120	4.7244	0.0008	29	1.1417	0.005	2.0	0.079	5.5	0.217	0.0010	0.0016
410	50	1.9685	0.0005	130	5.1181	0.0010	31	1.2205	0.005	2.0	0.079	6.5	0.256	0.0010	0.0016
411	55	2.1654	0.0006	140	5.5118	0.0010	33	1.2992	0.005	2.0	0.079	6.5	0.256	0.0010	0.0016
412	60	2.3622	0.0006	150	5.9055	0.0010	35	1.3780	0.005	2.0	0.079	6.5	0.256	0.0010	0.0016
413	65	2.5591	0.0006	160	6.2992	0.0010	37	1.4567	0.005	2.0	0.079	6.5	0.256	0.0010	0.0016
414	70	2.7559	0.0006	180	7.0866	0.0010	42	1.6535	0.005	2.5	0.098	7.5	0.295	0.0010	0.0016
415	75	2.9528	0.0006	190	7.4803	0.0012	45	1.7717	0.005	2.5	0.098	7.5	0.295	0.0010	0.0016
416	80	3.1496	0.0006	200	7.8740	0.0012	48	1.8898	0.005	2.5	0.098	7.5	0.295	0.0012	0.0018
417	85	3.3465	0.0008	210	8.2677	0.0012	52	2.0472	0.005	3.0	0.118	9.5	0.374	0.0012	0.0018
418	90	3.5433	0.0008	225	8.8583	0.0012	54	2.1260	0.005	3.0	0.118	9.5	0.374	0.0012	0.0018
419	95	3.7402	0.0008	240	9.4488	0.0012	55	2.1654	0.010	3.0	0.118	9.5	0.374	0.0012	0.0018
420	100	3.9370	0.0008	250	9.8425	0.0012	58	2.2835	0.010	3.0	0.118	9.5	0.374	0.0012	0.0018
421	105	4.1339	0.0008	260	10.2362	0.0012	60	2.3622	0.010	3.0	0.118	9.5	0.374	0.0012	0.0018
422	110	4.3307	0.0008	280	11.0236	0.0016	65	2.5591	0.010	3.0	0.118	9.5	0.374	0.0012	0.0018
424	120	4.7244	0.0008	310	12.2047	0.0016	72	2.8346	0.010	4.0	0.157	12.0	0.472	0.0014	0.0020
426	130	5.1181	0.0010	340	13.3858	0.0016	78	3.0709	0.010	4.0	0.157	12.0	0.472	0.0014	0.0020
428	140	5.5118	0.0010	360	14.1732	0.0016	82	3.2283	0.010	4.0	0.157	12.0	0.472	0.0014	0.0020
430	150	5.9055	0.0010	380	14.9606	0.0024	85	3.3465	0.010	4.0	0.157	12.0	0.472	0.0014	0.0020
432	160	6.2992	0.0010	400	15.7480	0.0024	88	3.4646	0.010	4.0	0.157	12.0	0.472	0.0014	0.0020
434	170	6.6929	0.0010	420	16.5354	0.0024	92	3.6620	0.010	4.0	0.157	12.0	0.472	0.0014	0.0020
436	180	7.0866	0.0010	440	17.3228	0.0024	95	3.7402	0.010	5.0	0.197	15.0	0.591	0.0014	0.0020
438	190	7.4803	0.0012	460	18.1102	0.0024	98	3.8583	0.010	5.0	0.197	15.0	0.591	0.0014	0.0020
440	200	7.8740	0.0012	480	18.8976	0.0024	102	4.0157	0.010	5.0	0.197	15.0	0.591	0.0014	0.0020
444	220	8.6614	0.0012	540	21.2598	0.0024	115	4.5276	0.010	5.0	0.197	15.0	0.591	0.0018	0.0024
448	240	9.4488	0.0012	580	22.8346	0.0024	122	4.8031	0.010	5.0	0.197	15.0	0.591	0.0018	0.0024

Note 1—The corner radius or chamfer on bearings must clear the maximum fillet radius given in the table and provide for sufficient bearing area against the minimum shoulder on the shafts.

Conversion from metric to decimal inch dimensions is according to the formula 1 mm. = 0.0393700 inch.

ANNULAR BALL AND ROLLER BEARINGS

WIDE TYPE

Table 1—Regular Light Series

Bearing Number	Bore, Nominal, Mm.	Bore, Nominal, In.	Bore, Toler., In. +0.0000	Outside Diameter, Nominal, Mm.	Outside Diameter, Nominal, In.	Outside Diameter, Toler., In. +0.0000	Width, In., Nominal	Width, In., Toler. +0.000	Shaft and Housing Fillet Radii, Max., Mm.	Shaft and Housing Fillet Radii, Max., In.	Shoulder Height on Shaft, Min., Mm.	Shoulder Height on Shaft, Min., In.
5200	10	0.3937	—0.0004	30	1.1811	—0.0004	9/16	—0.005	0.6	0.024	2.5	0.098
5201	12	0.4724	0.0004	32	1.2598	0.0005	5/8	0.005	0.6	0.024	2.5	0.098
5202	15	0.5906	0.0004	35	1.3780	0.0005	5/8	0.005	0.6	0.024	2.5	0.098
5203	17	0.6693	0.0004	40	1.5748	0.0005	11/16	0.005	1.0	0.039	3.0	0.118
5204	20	0.7874	0.0004	47	1.8504	0.0005	13/16	0.005	1.0	0.039	3.0	0.118
5205	25	0.9843	0.0004	52	2.0472	0.0006	13/16	0.005	1.0	0.039	3.0	0.118
5206	30	1.1811	0.0004	62	2.4409	0.0006	15/16	0.005	1.0	0.039	3.0	0.118
5207	35	1.3780	0.0005	72	2.8346	0.0006	1 1/16	0.005	1.0	0.039	3.5	0.138
5208	40	1.5748	0.0005	80	3.1496	0.0006	1 3/16	0.005	1.0	0.039	3.5	0.138
5209	45	1.7717	0.0005	85	3.3465	0.0008	1 3/16	0.005	1.0	0.039	3.5	0.138
5210	50	1.9685	0.0005	90	3.5433	0.0008	1 3/16	0.005	1.0	0.039	3.5	0.138
5211	55	2.1654	0.0006	100	3.9370	0.0008	1 5/16	0.005	1.5	0.059	4.5	0.177
5212	60	2.3622	0.0006	110	4.3307	0.0008	1 7/16	0.005	1.5	0.059	4.5	0.177
5213	65	2.5591	0.0006	120	4.7244	0.0008	1 1/2	0.005	1.5	0.059	4.5	0.177
5214	70	2.7559	0.0006	125	4.9213	0.0010	1 9/16	0.005	1.5	0.059	4.5	0.177
5215	75	2.9528	0.0006	130	5.1181	0.0010	1 5/8	0.005	1.5	0.059	4.5	0.177
5216	80	3.1496	0.0006	140	5.5118	0.0010	1 3/4	0.005	2.0	0.079	5.0	0.197
5217	85	3.3465	0.0008	150	5.9055	0.0010	1 15/16	0.005	2.0	0.079	5.0	0.197
5218	90	3.5433	0.0008	160	6.2992	0.0010	2 1/16	0.005	2.0	0.079	5.0	0.197
5219	95	3.7402	0.0008	170	6.6929	0.0010	2 3/16	0.005	2.0	0.079	6.0	0.236
5220	100	3.9370	0.0008	180	7.0866	0.0010	2 3/8	0.005	2.0	0.079	6.0	0.236
5221	105	4.1339	0.0008	190	7.4803	0.0012	2 9/16	0.005	2.0	0.079	6.0	0.236
5222	110	4.3307	0.0008	200	7.8740	0.0012	2 3/4	0.005	2.0	0.079	6.0	0.236

Table 1A—Extended Light Series

Bearing Number	Bore, Nominal, Mm.	Bore, Nominal, In.	Bore, Toler., In. +0.0000	Outside Diameter, Nominal, Mm.	Outside Diameter, Nominal, In.	Outside Diameter, Toler., In. +0.0000	Width, In., Nominal	Width, In., Toler. +0.000	Shaft and Housing Fillet Radii, Max., Mm.	Shaft and Housing Fillet Radii, Max., In.	Shoulder Height on Shaft, Min., Mm.	Shoulder Height on Shaft, Min., In.
5224	120	4.7244	0.0008	215	8.4646	0.0012	3	0.005	2.0	0.079	6.0	0.236
5226	130	5.1181	0.0010	230	9.0551	0.0012	3 1/8	0.005	2.5	0.098	7.0	0.276
5228	140	5.5118	0.0010	250	9.8425	0.0012	3 1/4	0.005	2.5	0.098	7.0	0.276
5230	150	5.9055	0.0010	270	10.6299	0.0016	3 1/2	0.005	2.5	0.098	7.0	0.276
5232	160	6.2992	0.0010	290	11.4173	0.0016	3 7/8	0.005	2.5	0.098	7.0	0.276
5234	170	6.6929	0.0010	310	12.2047	0.0016	4 1/8	0.005	3.0	0.118	9.0	0.354
5236	180	7.0866	0.0010	320	12.5984	0.0016	4 1/4	0.005	3.0	0.118	9.0	0.354
5238	190	7.4803	0.0012	340	13.3858	0.0016	4 1/2	0.010	3.0	0.118	9.0	0.354
5240	200	7.8740	0.0012	360	14.1732	0.0016	4 3/4	0.010	3.0	0.118	9.0	0.354
5244	220	8.6614	0.0012	400	15.7480	0.0024	5 1/4	0.010	3.0	0.118	9.0	0.354
5248	240	9.4488	0.0012	440	17.3228	0.0024	5 3/4	0.010	3.0	0.118	9.0	0.354
5252	260	10.2362	0.0012	480	18.8976	0.0024	6 1/4	0.010	4.0	0.157	11.0	0.433
5256	280	11.0236	0.0016	500	19.6850	0.0024	6 1/2	0.010	4.0	0.157	11.0	0.433
5260	300	11.8110	0.0016	540	21.2598	0.0024	7	0.010	4.0	0.157	11.0	0.433
5264	320	12.5984	0.0016	580	22.8346	0.0024	7 1/2	0.010	4.0	0.157	11.0	0.433

NOTE.—The corner radius or chamfer on bearings must clear the maximum fillet radii given in the table and provide for sufficient bearing against the minimum shoulder on the shafts.

ANNULAR BALL AND ROLLER BEARINGS

WIDE TYPE

Table 2—Regular Medium Series

	Bore			Outside Diameter			Width, In.		Shaft and Housing Fillet Radii, Max.		Shoulder Height on Shaft, Min.	
	Nominal			Nominal								
Bearing Number	Mm.	In.	Toler., In. +0.0000	Mm.	In.	Toler., In. +0.0000	Nominal	Toler. +0.000	Mm.	In.	Mm.	In.
5300	10	0.3937	—0.0004	35	1.3780	—0.0005	$\frac{3}{4}$	—0.005	0.6	0.024	2.5	0.098
5301	12	0.4724	0.0004	37	1.4567	0.0005	$\frac{3}{4}$	0.005	1.0	0.039	3.0	0.118
5302	15	0.5906	0.0004	42	1.6535	0.0005	$\frac{3}{4}$	0.005	1.0	0.039	3.0	0.118
5303	17	0.6693	0.0004	47	1.8504	0.0005	$\frac{7}{8}$	0.005	1.0	0.039	3.0	0.118
5304	20	0.7874	0.0004	52	2.0472	0.0006	$\frac{7}{8}$	0.005	1.0	0.039	3.5	0.138
5305	25	0.9843	0.0004	62	2.4409	0.0006	1	0.005	1.0	0.039	3.5	0.138
5306	30	1.1811	0.0004	72	2.8346	0.0006	$1\frac{3}{16}$	0.005	1.0	0.039	3.5	0.138
5307	35	1.3780	0.0005	80	3.1496	0.0006	$1\frac{3}{8}$	0.005	1.5	0.059	4.5	0.177
5308	40	1.5748	0.0005	90	3.5433	0.0008	$1\frac{7}{16}$	0.005	1.5	0.059	4.5	0.177
5309	45	1.7717	0.0005	100	3.9370	0.0008	$1\frac{9}{16}$	0.005	1.5	0.059	4.5	0.177
5310	50	1.9685	0.0005	110	4.3307	0.0008	$1\frac{3}{4}$	0.005	2.0	0.079	5.0	0.197
5311	55	2.1654	0.0006	120	4.7244	0.0008	$1\frac{15}{16}$	0.005	2.0	0.079	5.0	0.197
5312	60	2.3622	0.0006	130	5.1181	0.0010	$2\frac{1}{8}$	0.005	2.0	0.079	6.0	0.236
5313	65	2.5591	0.0006	140	5.5118	0.0010	$2\frac{5}{16}$	0.005	2.0	0.079	6.0	0.236
5314	70	2.7559	0.0006	150	5.9055	0.0010	$2\frac{1}{2}$	0.005	2.0	0.079	6.0	0.236
5315	75	2.9528	0.0006	160	6.2996	0.0010	$2\frac{11}{16}$	0.005	2.0	0.079	6.0	0.236
5316	80	3.1496	0.0006	170	6.6929	0.0010	$2\frac{11}{16}$	0.005	2.0	0.079	6.0	0.236
5317	85	3.3465	0.0008	180	7.0866	0.0010	$2\frac{7}{8}$	0.005	2.5	0.098	7.0	0.276
5318	90	3.5433	0.0008	190	7.4803	0.0012	$2\frac{7}{8}$	0.005	2.5	0.098	7.0	0.276
5319	95	3.7402	0.0008	200	7.8740	0.0012	$3\frac{1}{16}$	0.005	2.5	0.098	7.0	0.276
5320	100	3.9370	0.0008	215	8.4646	0.0012	$3\frac{1}{4}$	0.005	2.5	0.098	7.0	0.276
5321	105	4.1339	0.0008	225	8.8583	0.0012	$3\frac{7}{16}$	0.005	2.5	0.098	7.0	0.276
5322	110	4.3307	0.0008	240	9.4488	0.0012	$3\frac{5}{8}$	0.005	2.5	0.098	7.0	0.276

Table 2A—Extended Medium Series

5324	120	4.7244	0.0008	260	10.2362	0.0012	$4\frac{1}{8}$	0.010	2.5	0.098	7.0	0.276
5326	130	5.1181	0.0010	280	11.0236	0.0016	$4\frac{3}{8}$	0.010	3.0	0.118	9.0	0.354
5328	140	5.5118	0.0010	300	11.8110	0.0016	$4\frac{1}{2}$	0.010	3.0	0.118	9.0	0.354
5330	150	5.9055	0.0010	320	12.5984	0.0016	$4\frac{7}{8}$	0.010	3.0	0.118	9.0	0.354
5332	160	6.2992	0.0010	340	13.3858	0.0016	$5\frac{1}{4}$	0.010	3.0	0.118	9.0	0.354
5334	170	6.6929	0.0010	360	14.1732	0.0016	$5\frac{1}{2}$	0.010	3.0	0.118	9.0	0.354
5336	180	7.0866	0.0010	380	14.9606	0.0024	$5\frac{3}{4}$	0.010	3.0	0.118	9.0	0.354
5338	190	7.4803	0.0012	400	15.7480	0.0024	6	0.010	4.0	0.157	11.0	0.433
5340	200	7.8740	0.0012	420	16.5354	0.0024	$6\frac{1}{2}$	0.010	4.0	0.157	11.0	0.433
5344	220	8.6614	0.0012	460	18.1102	0.0024	7	0.010	4.0	0.157	11.0	0.433
5348	240	9.4488	0.0012	500	19.6850	0.0024	$7\frac{1}{2}$	0.010	4.0	0.157	11.0	0.433
5352	260	10.2362	0.0012	540	21.2598	0.0024	8	0.010	5.0	0.197	14.0	0.551
5356	280	11.0236	0.0016	580	22.8346	0.0024	$8\frac{1}{2}$	0.010	5.0	0.197	14.0	0.551

Note.—The corner radius or chamfer on bearings must clear the maximum fillet radii given in the table and provide for sufficient bearing against the minimum shoulder on the shafts.

ANNULAR BALL AND ROLLER BEARINGS

WIDE TYPE

Table 3—Regular Heavy Series

Bearing Number	Bore			Outside Diameter			Width, In.		Shaft and Housing Fillet Radii, Max.		Shoulder Height on Shaft, Min.	
	Nominal		Toler., In. +0.0000	Nominal		Toler., In. +0.0000	Nominal	Toler. +0.000	Mm.	In.	Mm.	In.
	Mm.	In.		Mm.	In.							
5403	17	0.6693	—0.0004	62	2.4409	—0.0006	1 3/16	—0.005	1.0	0.039	4.5	0.177
5404	20	0.7874	0.0004	72	2.8346	0.0006	1 3/8	0.005	1.0	0.039	4.5	0.177
5405	25	0.9843	0.0004	80	3.1496	0.0006	1 3/8	0.005	1.5	0.059	5.0	0.197
5406	30	1.1811	0.0004	90	3.5433	0.0008	1 9/16	0.005	1.5	0.059	5.0	0.197
5407	35	1.3780	0.0005	100	3.9370	0.0008	1 3/4	0.005	1.5	0.059	5.0	0.197
5408	40	1.5748	0.0005	110	4.3307	0.0008	1 15/16	0.005	2.0	0.079	5.0	0.197
5409	45	1.7717	0.0005	120	4.7244	0.0008	2 1/8	0.005	2.0	0.079	5.0	0.197
5410	50	1.9685	0.0005	130	5.1181	0.0010	2 5/16	0.005	2.0	0.079	6.5	0.256
5411	55	2.1654	0.0006	140	5.5118	0.0010	2 1/2	0.005	2.0	0.079	6.5	0.256
5412	60	2.3622	0.0006	150	5.9055	0.0010	2 5/8	0.005	2.0	0.079	6.5	0.256
5413	65	2.5591	0.0006	160	6.2992	0.0010	2 13/16	0.005	2.0	0.079	6.5	0.256
5414	70	2.7559	0.0006	180	7.0866	0.0010	3 1/8	0.005	2.5	0.098	7.5	0.295
5415	75	2.9528	0.0006	190	7.4803	0.0012	3 1/4	0.005	2.5	0.098	7.5	0.295
5416	80	3.1496	0.0006	200	7.8740	0.0012	3 7/16	0.005	2.5	0.098	7.5	0.295
5417	85	3.3465	0.0008	210	8.2677	0.0012	3 5/8	0.005	3.0	0.118	9.5	0.374
5418	90	3.5433	0.0008	225	8.8583	0.0012	3 7/8	0.005	3.0	0.118	9.5	0.374
5419	95	3.7402	0.0008	240	9.4488	0.0012	4 3/16	0.010	3.0	0.118	9.5	0.374
5420	100	3.9370	0.0008	250	9.8425	0.0012	4 1/2	0.010	3.0	0.118	9.5	0.374

NOTE.—The corner radius or chamfer on bearings must clear the maximum fillet radii given in the table and provide for sufficient bearing against the minimum shoulder on the shafts.

have these tables complete, the tolerances in the tables applying alike to both roller and ball bearings. On approval of the report by the Society the present S.A.E. table, which will continue to include only the regular series of sizes, will be reprinted to correspond to the accompanying tables with respect to detail dimensions.

Electrical Equipment Division

PERSONNEL

A. R. Lewellen, *Chairman*	Chevrolet Motor Co.
D. M. Pierson, *Vice-Chairman*	Chrysler Corp.
Azel Ames	Kerite Insulated Wire & Cable Co. Inc.
A. K. Brumbaugh	White Motor Co.
D. S. Cole	Leece-Neville Co.
W. S. Haggott	Packard Electric Co.
L M. Kanters	Waukesha Motor Co.
T. L. Lee	North East Electric Co.
L. E. Lighton	Electric Storage Battery Co.
L. O. Parker	Delco-Remy Corp.
E K. Schadt	Cadillac Motor Car Co.
T. E. Wagar	Studebaker Corp.

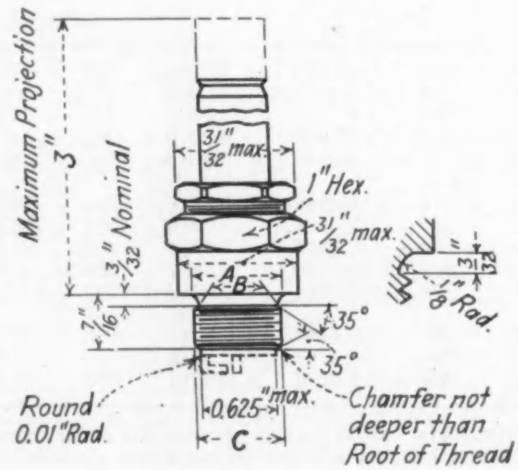
Metric Spark-Plugs

(*Proposed Revision of S.A.E. Recommended Practice*)

The Sub-Division on Metric Spark-Plugs took under consideration the criticism relative to the change made in the length of maximum reach to the bottom of the skirt of the metric spark-plug as approved last June. This change shortened the 9/16-in. dimension to ½ in. which, in the opinion of many, made it unsuitable for passenger-car engine-use. After considerable discussion it was found that a quantity of plugs, both of the 9/16 in. and ½ in. reach, were used.

In view of this fact it was considered advisable to eliminate this dimension entirely from the specifications for the present, to specify length of thread as 7/16 in. and dot in the lower end of the plug as in the illustration, and omit the dimension "Maximum reach ⅝ in."

The Division is of the opinion that this specification should be allowed to stand as illustrated for the next 6 months or a year to determine which length of plug finds the greater use in engine design, after which a definite dimension will then be set.



The Division, therefore, recommends that the changes as shown in the accompanying illustration be approved; all other tables of dimensions and tolerances on metric spark-plugs to stand as in the present S.A.E. Recommended Practice, p. 29 of Supplement to the 1929 edition of the S.A.E. HANDBOOK.

Spark-Plug Tests

(Proposed Cancellation of S.A.E. Recommended Practice)

At the meeting of the Sub-Division on Metric Spark-Plugs on Dec. 19 the question of the value of the present S.A.E. Recommended Practice on Spark Plug Tests, p. 143 of the 1929 edition of the S.A.E. HANDBOOK was taken up with the engine and spark-plug manufacturers present. As the result of the discussion on this matter, the recommendation was made that the specifications be cancelled as being inadequate. Opinion was also expressed that attempting to develop new specifications for such tests was not feasible.

The subject has therefore been submitted to the Electrical Equipment Division and the Engine Division by letter-ballot and is herewith submitted to the Standards Committee for cancellation.

This is submitted to the Standards Committee subject to letter-ballot approval by the Electrical Equipment Division.

Starting-Switch Location

(Proposed Cancellation of S.A.E. Recommended Practice)

The Division recommends the cancellation of the present S.A.E. Recommended Practice on Starting-Switch Location, p. 153 of the 1929 edition of the S.A.E. HANDBOOK as it is now considered that the specification is not definite enough to be of any value.

This is submitted to the Standards Committee subject to letter-ballot approval by the Electrical Equipment Division.

Temperature Test of Insulating Materials

(Proposed Cancellation of S.A.E. Standard)

This specification was adopted in 1914 and since that time sufficient changes have been made in the insulating-material field to render it valueless. The Standards Committee is therefore requested to approve the cancellation of the S.A.E. Standard on Temperature Test of Insulating Materials, p. 158 of the 1929 edition of the S.A.E. HANDBOOK.

This is submitted to the Standards Committee subject to letter-ballot approval by the Electrical Equipment Division.

Starting-Motor Cable

(Proposed Cancellation of S.A.E. Standard)

The S.A.E. Standard on Starting-Motor Cables, p. 163 of the 1929 edition of the S.A.E. HANDBOOK was adopted in 1918 and covers the maximum cable-length for three cable-sizes based on an allowable voltage-drop. The Division feels that no purpose is served by this standard and therefore recommends that it be cancelled.

This is submitted to the Standards Committee subject to letter-ballot approval by the Electrical Equipment Division.

Rubber-Bushing Holes

(Proposed Cancellation of S.A.E. Recommended Practice)

Since this specification was adopted in 1922 new specifications have been developed for rubber bushings which render the necking of holes unnecessary. The Division therefore recommends that the S.A.E. Recommended Practice on Rubber-Bushing Holes, appearing on p. 167 of the 1929 edition of the S.A.E. HANDBOOK, be cancelled.

This is submitted to the Standards Committee subject to letter-ballot approval by the Electrical Equipment Division.

Iron and Steel Division

PERSONNEL

J. M. Watson, *Chairman*	Hupp Motor Car Corp.
A. H. D'Arcambal, *Vice-Chairman*	Pratt & Whitney Co.
J. R. Adams	Midvale Co.
R. J. Allen	Rolls Royce of America, Inc.
A. L. Boeghold	General Motors Corp.
Henry Chandler	Vanadium Corp. of America
J. D. Cutter	Fafnir Bearing Co.
B. H. DeLong	Carpenter Steel Co.
F. P. Gilligan	Henry Souther Engineering Corp.
H. W. Graham	Jones & Laughlin Steel Corp.
H. L. Greene	Willys-Overland, Inc.
W. G. Hildorf	Timken Steel & Tube Co.
E. J. Janitzky	Illinois Steel Co.
J. A. Mathews	Crucible Steel Co. of America
W. C. Peterson	Donner Steel Co.
E. A. Portz	Central Alloy Steel Corp.
S. P. Rockwell	Stanley P. Rockwell Co.
R. B. Schenck	Buick Motor Co.
Ralph R. Teetor	Perfect Circle Co.
H. P. Tiemann	Carnegie Steel Co.
E. W. Upham	Chrysler Motors
T. H. Wickenden	International Nickel Co.
Henry Wysor	Bethlehem Steel Co.
O. B. Zimmerman	International Harvester Co.

Heat-Treatment Definitions

(Proposed Revision of S.A.E. General Information)

The present General Information on Heat-Treatment Definitions printed in the 1929 edition of the S.A.E. HANDBOOK commencing on p. 394, is submitted for revision in accordance with a revised report received from the Joint Committee on Heat-Treatment Definitions. These definitions were originally adopted by the Society in June, 1927, but the Joint Committee has felt that in view of a number of comments and criticisms received since then, the Definitions should be revised.

The Joint Committee is made up of representatives of the three Societies, the American Society of Steel Treaters, the American Society for Testing Materials, and the Society of Automotive Engineers, and they submitted to the Division the following changes. These changes were in turn submitted to the Iron and Steel Division for approval by letter-ballot and are hereby presented to the Standards Committee as a revision of the existing data.

The proposed changes are as follows:

Quenching.—This definition was changed from "immersion" to "cool."

Annealing.—The order of the Notes (*a*) to (*e*) inclusive were changed from the present order shown in the S.A.E. HANDBOOK to 5, 1, 2, 3, 4.

The following three definitions have been added under the general heading "Annealing":

Full Annealing.—Heating iron-base alloys above the critical-temperature range, holding above that range for a proper period of time followed by slow cooling through the range.

Note.—The annealing temperature is generally about 100 deg. fahr. above the upper limit of the critical-temperature range, and the time of holding is usually not less than 1 hr. for each inch of section of the heaviest objects being treated. The objects being treated are ordinarily allowed to cool slowly in the furnace. They may, however, be removed from the furnace and cooled in some medium

that will prolong the time of cooling as compared to unrestricted cooling in the air.

Process Annealing.—Heating iron-base alloys to a temperature below or close to the lower limit of the critical-temperature range followed by cooling as desired.

Note.—This heat-treatment is commonly applied in the sheet and wire industries and the temperatures generally used are from 1020 to 1200 deg. fahr.

Patenting.—Heating iron-base alloys above the critical-temperature range followed by cooling to below that range in air or in molten lead that is maintained at a temperature of about 700 deg. fahr.

Note.—This treatment is applied in the wire industry as a finishing treatment or especially in the case of eutectoid steel as a treatment previous to further wire drawing. Its purpose is to produce a sorbitic structure.

The definition of Tempering has been changed by the addition of the word "desired" to read as follows:

Tempering. (Also termed Drawing).—Reheating after hardening to some temperature below the critical-temperature range followed by any desired rate of cooling.

Note.—(*a*) Although the terms "tempering" and "drawing" are practically synonymous as used in commercial practice, the term "tempering" is preferred.

(*b*) Tempering, meaning the operation of hardening followed by reheating, is a usage that is illogical and confusing in the present state of the art of heat treating and should be discouraged.

The definition of Graphitizing has been changed by the addition of the word "gray" and the new definition reads as follows:

Graphitizing.—Graphitizing is a type of annealing for gray cast iron whereby some or all of the combined carbon is transformed to free or uncombined carbon.

The note under the definition of Carburizing has been changed by the substitution of the word "incorrect" for "undesirable," the definition with the revised note reading as follows:

Carburizing. (*Cementation*).—Adding carbon to iron-base alloys by heating the metal below its melting point in contact with carbonaceous material.

Note.—The term "carbonizing" used in this sense is incorrect so its use should be discouraged.

The note under the definition of Case Hardening has been amended to read as follows, changing the words "these definitions" to "the terms" and the addition of "case and core."

Note.—The terms "case" and "core" refer to both case hardening and carburizing.

Lighting Division

PERSONNEL

C. A. Michel, *Chairman*	Guide Lamp Corp
R. N. Falge, *Vice-Chairman*	General Motors Corp.
Clyde C. Bohner	Tung-Sol Lamp Works
R. E. Carlson	Westinghouse Lamp Co.
A. W Devine.	Massachusetts Registry of Motor Vehicles
H. C. Doane	Buick Motor Co.
G. P. Doll	Thos. J. Corcoran Lamp Co.
W S. Hadaway	Edison Lamp Works of the General Electric Co.
C. L. Holm	Ford Motor Co.
R. W. Johnson	John W. Brown Mfg. Co.
W. M. Johnson	National Lamp Works of the General Electric Co.
A. R. Lewellen	Chevrolet Motor Co.
A. L. Martinek	C. M. Hall Lamp Co.
D. M. Pierson	Chrysler Corp.
W. F. Thoms	Allied Products Corp.
T. E. Wagar	Studebaker Corp.

Laboratory Tests of Optical Characteristics of Electric Head-Lamps for Motor-Vehicles

(*Proposed S.A.E. Standard Based on a Revision of Present S.A.E. Recommended Practice*)

The present specifications for Laboratory Tests of Optical Characteristics of Electric Head-Lamps for Motor-Vehicles, p. 120 in the 1929 edition of the S.A.E. HANDBOOK, have, since coming into general use, necessitated some slight revision in the matter of tolerances. The Electrical Equipment Division, acting on the report submitted by the joint sub-committee of the Illuminating Engineering Society and the Society has approved a revision of these specifications as reprinted herewith and recommends that the Standards Committee approve these as S.A.E. Standard to supplant the present Recommended Practice on the same subject.

Specifications for Laboratory Tests of Optical Characteristics of Electric Head-Lamps for Motor-Vehicles

Definitions.—Head-lamp, a lighting device used on the front of a vehicle primarily to provide general illumination ahead of the vehicle. Spotlamps and similar specialized lighting-devices are not classified as head-lamps unless they are used or intended to be used to perform the ordinary functions of head-lamps as defined above.

Dual-beam equipment, head-lamps or similar devices arranged so as to permit the driver of the vehicle to use either of two distributions of light on the road.

Scope of Specification.—This specification is intended to cover the optical performance of present conventional types of head-lamp.

Samples for Test.—Sample head-lamps representative of the type as regularly manufactured and marked shall be submitted to the laboratory for test. Such samples shall include all accessory equipment peculiar to the device and necessary to operate it in its normal manner, except that the socket sleeve may be omitted from the reflector. The samples shall be accompanied by printed, typewritten or written instructions for adjustment, which will enable the laboratory operator to determine when the light source is located in the position it is designed to have and to make any other adjustments necessary to obtain proper results from the lamps. The laboratory report shall include a copy of these instructions for adjustments.

Incandescent Lamps.—The incandescent lamps used in the tests, unless otherwise specified, shall be supplied by the laboratory. They shall be of standard manufacture of either 21 or 32 cp.[1] as specified by the applicant. They shall be of a construction approved for this purpose by the Bureau of Standards. The lamps shall be such as will give their rated candlepower when operated approximately at their rated efficiency. The results of photometric tests shall be based on the lamps operated at rated candlepower while the tests are being made.

Set-Up for Testing.—The laboratory shall be provided with facilities for making accurate measurements in accordance with established laboratory practices. The head-lamps shall be tested singly or in pairs as used and under conditions of installation similar to those in general use on motor-vehicles. A photometer shall be set up not less than 60 nor more than 100 ft. from the head-lamps. When lamps are tested in pairs, the vertical planes through the axes of the lamps shall be parallel. If a testing distance of 100 ft. is taken, the lamps shall be 28 in. apart from center to center; if a shorter testing-distance is taken, the distance between head-lamps shall be proportionately reduced.

Photometric Test-Points.—To assist in locating the test points, the following nomenclature has been adopted: The line formed by the intersection of the median vertical-plane parallel to the lamp axes and the test-screen shall be desig-

[1] The trend toward wider and deeper beams has created a definite need for the higher candlepower incandescent lamps. The laws or regulations of certain States, however, establish candlepower limits for the lamps, bulbs, to be used. Tests made in connection with approval in such States should be made with lamps of the candlepower specified.

nated as V. The line formed by the intersection of the horizontal plane through the head-lamp centers and the test-screen shall be designated as H. The point at the intersection of these two lines shall be designated as H-V. The other points on the screen shall be designated by similar symbols to indicate the number of degrees of arc above or below H and the number of degrees of arc to the left or right of V, for example: $4D$-$3L$ is a point 4 deg. below H and 3 deg. to the left of V and $1U$-V is a point 1 deg. above H in the median vertical-plane.

The intensity of the combined light from the pair of head-lamps shall be measured unless the samples are not intended to be used in pairs. In the latter case the samples shall be tested as used. If the beam is symmetrical about the median vertical-plane, measurements shall be made at 1-deg. intervals over an area extending from the center V to a line 12 deg. to the left $12L$ and from a line 2 deg. above the head-lamp level $2U$ to a line 6 deg. below the head-lamp level $6D$. Measurements shall be made at any additional points for which limitations are set. If the beam is not symmetrical about the median vertical-plane, measurements shall also be taken in similar manner to the right of the median vertical-plane.

Focal Adjustments of Incandescent Lamps.—Head-lamps shall be adjusted for the normal or designed position of the light source in accordance with the instructions of the applicant. Complete distribution tests shall be made with the filament of the incandescent lamp in each of five positions as follows:

(1) Designed position
(2) 0.06 in. above designed position
(3) 0.06 in. below designed position
(4) 0.06 in. ahead of designed position
(5) 0.06 in. behind designed position

For dual-beam head-lamps, distribution tests shall be made on both the upper and the lower beam with the light source in each of the specified positions.

Aiming Adjustment of Head-Lamps.—Each pair of head-lamps shall be aimed once for each position of the light source. Aiming adjustments for dual-beam head-lamps shall be made for upper beams only.

Beam-Candlepower Requirements.—While complete distribution tests are to be made for all filament positions, as specified under Focal Adjustments, compliance with the specifications shall be determined by candlepower limits at certain points or on certain lines.

Specifications for Laboratory Tests of Head-lamps

The values that follow are based upon the performance of the dual-beam head-lamps with no loading allowance for the upper beam or of fixed-beam head-lamps with the proper allowance. The single beam from the fixed-beam head-lamps and the upper beam from dual-beam head-lamps shall comply with the limitations printed at the bottom of the other column.

	Candlepower Requirements	
	Filament at Designed Position [2]	For Out-of-Designed Filament Positions Specified [2]
Point H-V (Reference Point)	2,000 for 21-cp. lamps; 3,000 for 32-cp. lamps	2,000 for 21-cp. lamps; 3,000 for 32-cp. lamps
Point $1U$-V	400 to 2,000	400 to 2,000
Point $1U$-$4L$	800 or less	800 or less
Line $1D$-$3L$ to $3R$	7,500 or more	6,000 or more
Line $2D$-$6L$ to $6R$	4.000 or more	3,200 or more
Line $3D$-$9L$ to $9R$	2,000 or more	1,600 or more
Maximum-permissible beam-candlepower, any point	50,000	50,000
Point of maximum beam-candlepower not lower than	$2D$	$2\frac{1}{2}D$

[2] A tolerance of $\pm\frac{1}{2}$ deg. vertically is permitted at all points below H-V.

The lower beam from dual-beam head-lamps shall comply with the following limitations:

	Candlepower Requirements	
	Filament at Designed Position [3]	For Out-of-Designed Filament Positions Specified [3]
Line $1U$ from V to left	800 or less	800 or less
Line H from V to left	1,500 or less	1,500 or less
Line $1D$ from V to left	2,400 or less	2,400 or less
Point $2D$-V	6,000 or less	6,000 or less
Point $3D$-V	5,000 or more	4,000 or more
Line $4D$ or $5D$ or $6D$, $12L$ to $12R$, only one required	1,000 or more	1,000 or more
Maximum-permissible beam-candlepower, any point	25,000 for 21-cp. lamps; 37,500 for 32-cp. lamps	30,000 for 21-cp. lamps; 45,000 for 32-cp. lamps

[3] A tolerance of $\pm\frac{1}{2}$ deg. vertically is permitted at all points below H-V.

The lower beam as provided by either one or two auxiliary driving-lamps shall comply with the following limitations:

	Candlepower Requirements	
	Filament at Designed Position [4]	For Out-of-Designed Filament Positions Specified [4]
Line $1U$ from V to left	800 or less	800 or less
Point H-V	1,000 for 21-cp. lamps; 1,500 for 32-cp. lamps	1,000 for 21-cp. lamps; 1,500 for 32-cp. lamps
Line $1D$ from V to left	2,400 or less	2,400 or less
Point $2D$-V	6,000 or less	6,000 or less
Point $3D$-V	5,000 or more	4,000 or more
Line $4D$ or $5D$ or $6D$, $12L$ to $12R$, only one required	1,000 or more	1,000 or more
Maximum-permissible beam-candlepower, any point	25,000 for 21-cp. lamps; 37,500 for 32-cp. lamps	30,000 for 21-cp. lamps; 45,000 for 32-cp. lamps

[4] A tolerance of $\pm\frac{1}{2}$ deg. vertically is permitted at all points below H-V.

It is desirable that all classes of head-lamp meet the candlepower requirements for all five filament-positions but compliance is mandatory for the several classes of equipment as follows:

Head-lamps equipped with no focus-adjusting mechanism:
Filament position:

(1) Designed position
(2) 0.06 in. above designed position
(3) 0.06 in. below designed position
(4) 0.06 in. ahead of designed position
(5) 0.06 in. behind designed position

Head-lamps equipped with a horizontal focus-adjusting mechanism only:
Filament position:

(1) Designed position
(2) 0.06 in. above designed position
(3) 0.06 in. below designed position

Head-lamps equipped with a vertical focus-adjusting mechanism, whether or not they are also equipped with a horizontal focus-adjusting mechanism:
Filament position:

(1) Designed position
(2) 0.06 in. above designed position

(3) 0.06 in. below designed position
(4) 0.06 in. ahead of designed position
(5) 0.06 in. behind designed position
(6) At the limits of vertical motion if such limits exceed 0.06 in.

Laboratory Tests of Optical Characteristics of Tail-Lamps for Motor-Vehicles

(Proposed S.A.E. Standard Replacing Present S.A.E. Recommended Practice)

Similarly acting upon that part of the joint sub-committee's report dealing with tail-lamps, the Divison approved the splitting of these specifications into two parts, namely, one governing the laboratory tests of optical characteristics and the other tail-lamp construction. The Standards Committee is requested to approve as S.A.E. Standard the following specifications to take the place of the present S.A.E. Recommended Practice on Tail-Lamps, p. 128 of 1929 edition of S.A.E. HANDBOOK.

Specifications for Laboratory Tests of Optical Characteristics of Tail-Lamps for Motor-Vehicles

Definition.—The tail-lamp is a device designed to indicate the rear end of a vehicle by a red light and also to illuminate the license or registration-number plate.

Scope of Specification.—This specification is limited to present conventional types of tail-lamps equipped with electric incandescent lamps and designed for opaque license-plates. It covers only the optical performance of such devices.

Samples for Test.—Sample tail-lamps submitted for laboratory test shall be representative of the devices as regularly manufactured and marketed. Each sample shall include the license-plate holder and any other accessory equipment peculiar to the device and necessary to operate it in its normal manner. The sample shall also be accompanied by any instructions necessary for determining its normal position when mounted on the vehicle.

Set-Up for Testing.—The laboratory shall be equipped with all facilities necessary to make accurate photometric measurements in accordance with established laboratory-practices.

Unless otherwise specified, the incandescent lamps used in tail-lamp tests shall be supplied by the laboratory. They shall be representative of standard 3-cp. 6 to 8-volt incandescent lamps in regular production for automotive service. They shall be selected for accuracy in accordance with specifications approved by the Bureau of Standards. They shall be operated at 3 mean spherical-cp. during the tests. Where special incandescent lamps are specified, such lamps shall be submitted with the device and the same or similar lamps used in the tests and operated at their rated mean spherical-candlepowers.

All beam-candlepower measurements shall be made with the center of light at a distance of 4 ft. from the photometer screen. In measuring distances and angles, the incandescent filament shall be taken as the center of light. [This paragraph is to be revised in accordance with agreement between W. F. Little and R. N. Falge.]

All foot-candle measurements shall be made on a rectangular test-plate of clean white blotting paper mounted on the license-plate holder in the position ordinarily taken by the license plate. This test-plate shall be at least 6 x 16 in. The face of the test-plate shall be 1/16 in. from the face of the license-plate holder.

Eight test-stations shall be located on the face of the test-plate. These test-stations shall be arranged in two horizontal rows and in four vertical rows. The horizontal rows shall be 1 and 5 in. respectively from the centers of the bolt slots. The vertical rows shall be 1 and 7 in. respectively to the left and right of the vertical center-line of the plate.

General Optical Requirements.—The tail-lamp assembly shall be designed so that, when properly mounted on a car on a level surface, the plane of the license plate shall not make an angle greater than 15 deg. with the vertical.

The tail-lamp and license-plate holder shall bear such a relation to each other that at no point on the plate will the major portion of the light incident at that point make an angle of less than 8 deg. to the plane of the plate.

The license-plate holder shall be designed and constructed so as to provide a substantially plane surface on which to mount the plate.

The license-plate window through which the light passes to the license plate shall be of such size and design as to permit the light to reach all parts of the license plate. The shadow cast by the edge of the license-plate window shall clear the far edge of the license plate, as represented by a line 6 in. from the bolt centers, by at least 1 in., measured perpendicularly to the plane of the plate.

Photometric-Test Requirements.—On a horizontal line through the light source and parallel to the longitudinal axis of a car the red light shall have an intensity of at least 0.10 cp.

In all directions within 30 deg. to this line and above the horizontal there shall be at least 0.05 cp. In no direction above the horizontal shall there be more than 5.0 cp.

The illumination at each of the eight stations on the test-plate shall be at least 0.5 ft. candle. The ratio of maximum illumination to minimum illumination shall not exceed 30 to 1. The average of the two highest and the two lowest illumination values recorded at the eight test-stations shall be taken as such maximum and minimum values respectively.

Tail-Lamp Construction

(Proposed S.A.E. Recommended Practice)

The following specification is submitted for the consideration of the Standards Committee as an S.A.E. Recommended Practice.

Tail-Lamp Construction

Tail-Lamp.—A device designed to indicate the rear end of a vehicle by a red light and also to illuminate the license plate.

(1) All rear lamps, including license-plate holders and supporting brackets, shall, through suitable design and choice of materials, be of such strength and rigidity as to insure proper illumination of the license plate under all ordinary conditions of service.

(2) The name of the rear lamp shall be prominently and permanently marked on the lamp body and also on the license-plate holder if the latter is detachable.

(3) The rear lamp shall be substantially weather and dust proof. It should also be constructed in such manner as to withstand the vibration normally encountered in service.

(4) The license-plate window in the rear lamp through which light passes to the license plate shall be covered with substantially colorless glass or other material of suitable strength and permanency.

(5) When tail-lamps are designed to center the license plate above or below the lamp and in which provision is made for bolting the license plate to the license-plate holder, the license-plate holder shall be provided with standard bolt-slots in the license-plate holder as shown in the S.A.E. specifications for license-plate bracket-slots.

(6) All license plates shall be provided with four standard round bolt holes, two at the top and two

at the bottom, as shown in the S.A.E. specifications for license-plate holes.

Tail and Signal Lamp Mountings

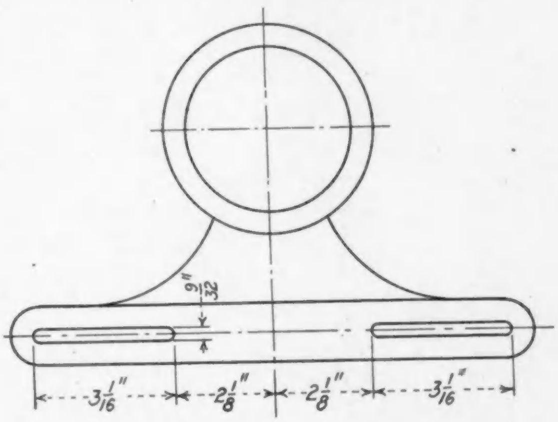
(Proposed Change from S.A.E. Recommended Practice to S.A.E. Standard)

The Division recommends that the present S.A.E. Recommended Practice on Tail and Signal Lamp Mountings, p. 128 of the 1929 edition of the S.A.E. HANDBOOK, be indicated therein as reapproved as of January, 1930, and made an S.A.E. Standard in view of the fact that this mounting is now in very general use and is accepted as a standard mounting.

License-Plate Bracket-Slots

(Proposed Revision of S.A.E. Standard)

In connection with the work of the Sub-Division on Tail Lamps and the joint committee of the Illuminating Engineering Society and the Society on this subject, the question of centering license plates on license-plate holders for the necessary illumination was taken up. To provide a license-plate holder to accomplish this and to conform to present practice for such a device, the Division recommends the revision of the present S.A.E. Standard on License-Plate Bracket-Slots in accordance with accompanying drawing. No change other than the lengthening of the slots by a 1/16 in. is made by this revision.



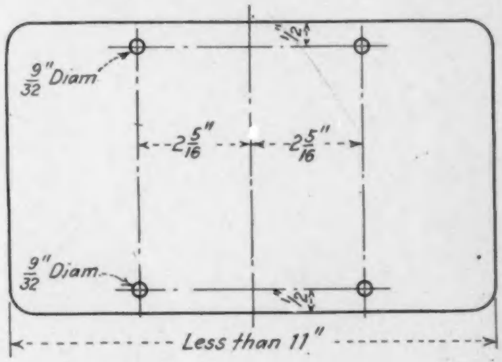
License-Plate Holes

(Proposed S.A.E. Standard)

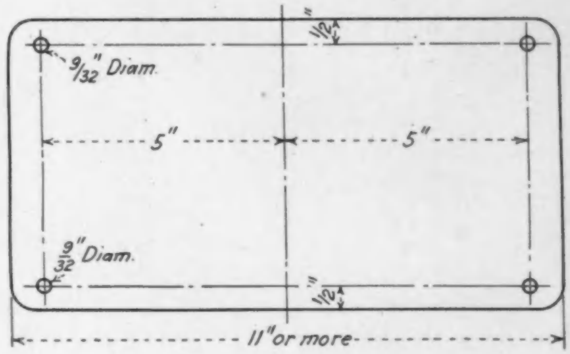
To accomplish the centering of license plates on brackets having the recommended slots, the Division has approved the accompanying specifications for location of license-plate holes and has requested that the Standards Department take this matter up with all of the States in an endeavor to obtain adherence to these specifications. The proposed standard calls for two holes at the top and two holes at the bottom of each license plate; one set of dimensions for the location of holes in plates less than 11 in. in length and another set for the location of holes in plates over 11 in. in length. Plates so placed will automatically center on the proposed S.A.E. Standard for license-plate bracket-holders.

License-Plate Holes

License plates less than 11 in. in length should have four holes located as illustrated.



License plates over 11 in. in length should have holes located as illustrated.



Motorboat Division

PERSONNEL

Leonard Ochtman, Jr., *Chairman*	Elco Works, Electric Boat Co.
C. A. Carlson	Remington Oil Engine Co.
N. E. Donnelly	Dawn Boat Corp.
H. E. Fromm	Chrysler Sales Corp.
S. Clyde Kyle	American Car & Foundry Co.
H. R. Sutphen	Submarine Boat Corp.
Jos. Van Blerck	Van Blerck Motors, Inc.

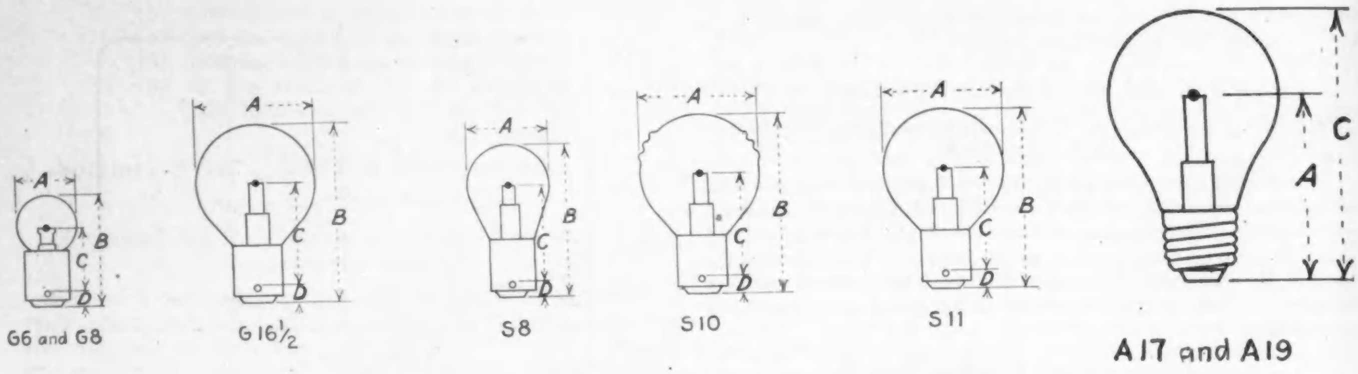
Motorboat Electric Incandescent Lamps

(Proposed S.A.E. Recommended Practice)

Considerable difficulty has been experienced in the past in the matter of obtaining replacements of motorboat incandescent lamps because of the lack of standardization of the lamps and the varying practices of motorboat manufacturers in the various types used.

In an endeavor to determine a suitable series of incandescent lamp sizes in the three voltages used in the motorboat lighting system, the Motorboat Division undertook a study of the question with representatives of the lamp manufacturers.

As a result, the table of lamps for motorboat use printed on p. 26 has been worked out and this is being submitted to the Standards Committee for approval, subject to approval or revision prior to the Standards Committee meeting by the Division.



MOTORBOAT ELECTRIC INCANDESCENT LAMPS

Bayonet Base Lamps

Candle Power	Voltage	Maximum Amperage	Type of Base[1]	Bulb Type	A Bulb Diameter, In.	B Maximum Over-All Length, In.	C Light-Center Length, In.[2]	D
3[a]	6–8	0.75	D. C	G-6	3/4	1 7/16	3/4	0.320
3[a]	12–16	0.50	D. C.	G-6	3/4	1 7/16	3/4	0.320
3[a]	32	0.15	D. C.	G-6	3/4	1 7/16	3/4	0.320
6	6–8	1.25	D. C.	G-6	3/4	1 7/16	3/4	0.320
6	12–16	0.75	D. C.	G-6	3/4	1 7/16	3/4	0.320
6[a]	32	0.30	D. C.	G-8	1	1 3/4	7/8	0.320
15[a]	6–8	2.00	D. C.	S-8	1	2	1 1/8	0.320
15[a]	12–16	1.25	D. C.	S-8	1	2	1 1/8	0.320
21[a]	6–8	3.00	D. C.	S-10[b]	1 1/4	2 3/8	1 1/4	0.320
21[a]	12–16	1.50	D. C.	S-10[b]	1 1/4	2 3/8	1 1/4	0.320
50	6–8	7.00	D. C.	S-11	1 3/8	2 3/8	1 1/4	0.320
50[a]	12–16	3.50	D. C.	S-11	1 3/8	2 3/8	1 1/4	0.320
Amps.								
3.5[a]	32	4.25	D. C.	G-16 1/2	2 1/16	3	1 11/16[c]	0.320
3.5	32	4.25	D. C.	G-16 1/2	2 1/16	3	1 1/4	0.320
Medium Screw Base Lamps[3]								
Watts								
10[a]	6	2.00	Med.	A-17	2 1/8	3 5/8	2 3/8	
15[a]	12	1.50	Med.	A-17	2 1/8	3 5/8	2 3/8	
15[a]	32	0.60	Med.	A-17	2 1/8	3 5/8	2 3/8	
25	32	1.00	Med.	A-19	2 3/8	3 15/16	2 1/2	

[1] D. C.—S.A.E. Standard double contact bayonet base; Med.—Medium screw type base.
[2] Light-center length and axial-alignment tolerances for headlight lamps are +3/64 in.
[3] The 10, 15 and 25-watt lamps are usually made with inside frosted bulbs only.
[a] Indicates most general use.
[b] The corrugations on electric incandescent lamps for headlight service shall be of sufficient depth to break up the filament image.
[c] Also made with 1¼-in. light center length.

Non-Ferrous Metals Division

PERSONNEL

Zay Jeffries, *Chairman*	Aluminum Co. of America
C. W. Simpson, *Vice-Chairman*	White Motor Co.
R. J. Allen	Rolls Royce of America, Inc.
W. H. Bassett	American Brass Co.
C. H. Calkins	Baush Machine Tool Co.
D. L. Colwell	Stewart Die Castings Corp.
H. R. Corse	Lumen Bearing Co.
W. A. Cowan	National Lead Co.
P. V. Faragher	Aluminum Co. of America
W. H. Graves	Packard Motor Car Co.
H. L. Greene	Willys-Overland, Inc.
J. B. Johnson	Materiel Division, Air Corps
R. R. Moore	Wright Aeronautical Corp.
H. C. Mougey	General Motors Corp.
A. L. Nelson	Bohn Aluminum & Brass Corp.
W. M. Peirce	New Jersey Zinc Co.
W. B. Price	Scovill Mfg. Co.
T. H. Wickenden	International Nickel Co.
H. M. Williams	Frigidaire Corp.

Chromium Plating

(Proposed S.A.E. General Information)

The Non-Ferrous Metals Division considers it advisable to add to the S.A.E. HANDBOOK data on the use of many of the non-ferrous metals, such as chromium and cadmium for plating and the use of the rarer metals in various parts of automobile construction.

Special committees are working at present on a variety of these subjects, and the information on chromium plating is submitted herewith by the Division as the first of a series for the approval of the Standards Committee as General Information to be published in the S.A.E. HANDBOOK.

Chromium Plating

Chromium has been plated by different experimenters since the middle of the last century, but it was not until after the work of H. R. Carveth and B. E. Curry[1] and G. J. Sargent[2] that the subject assumed commercial importance. A large number of experimenters have worked on this subject and many patents have been taken out covering particular details.

Practically all of the chromium plating done in this Country at present is from solutions of substantially the following composition, with sometimes minor additions. Chromic acid (CrO_3), 200 to 300 gm.; sulphates, calcu-

[1] See *Journal of Physical Chemistry*, May, 1905, p. 353; reprinted in *Transactions of the American Electrochemical Society*, vol. 7, p. 115.

[2] See *Transactions of the American Electrochemical Society*, vol. 37, p. 479.

lated as sulphuric acid, 1 per cent of the chromic acid and water to make 1000 cc. The current density and the temperature are the two important variables in the operation of the solution. Baths of this type usually produce a bright deposit over polished surfaces at a current density of 150 to 200 amp. per sq. ft. and a temperature of from 50 to 60 deg. cent. (122 to 140 deg. fahr.). Too high a current produces a burned or satin finish while too low a current gives a bluish plate or fails to cover. Variations in temperature produce like results.

Schneidewind[3] has shown that the relation of current

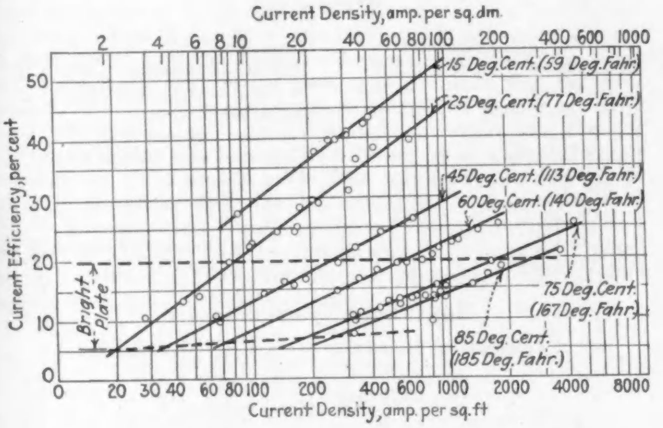


FIG. 1—CURRENT DENSITY-CURRENT EFFICIENCY RELATIONSHIPS

density, temperature, appearance of the plate and current efficiency can be plotted on a special semi-logarithmic paper as shown in Fig. 1.

The tank should be of steel, preferably low carbon, full annealed, although any material that the bath does not attack can be used, such as lead or glass-lined tanks. The steel tank can be used as an anode although it has the disadvantage that with steel anodes, the steel will slowly dissolve in the solution causing it to build up in iron oxide, thereby increasing the resistance and decreasing the throwing power. For this reason lead or lead-antimony alloy anodes are preferred by some platers. The distance between anodes and cathodes may vary from 1 to 12 in.

The composition of the bath is kept uniform by the addition of chromic acid, water and, if the chromic acid does not contain sufficient sulphates as impurities, sulphuric acid or soluble sulphates. Care should be taken that the ratio of chromic acid to sulphates, calculated as sulphuric acid, is always kept at 100 to 1.

Chromium is used both for appearance and prevention of corrosion and for resistance to wear. The chromium forms a tarnish-resisting coating over other metals or plates, such as copper or nickel. In general, thin films of chromuim are used for this purpose. The nickel and copper should each be at least 0.0004 in. thick, in which case a 3 to 5 min. plate of chromium is sufficient. The chromium can be applied directly to hardened steel or other surfaces to provide a very hard wear-resisting surface, since electrodeposited chromium is very hard. However, to secure best results when used in this manner the chromium should be applied in relatively thick layers and the metal on which the chromium is deposited should be as hard as possible.

The piece to be plated, whether steel, nickel-plated steel, copper or brass, is generally cleaned first in an alkali cleaner with or without the use of current. It is then rinsed in water, dipped in hydrochloric acid, rinsed and immediately placed in the chromium bath while still wet. If a polished chromium surface is required, the article should be polished and buffed before cleaning. Sometimes other methods of cleaning are used. When plating chromium over a nickel deposit some trouble may be experienced due to the nickel peeling. This is best overcome by plating a heavy deposit of nickel from a hot solution. When plating chromium over nickel, it is necessary to keep the hydrochloric acid dip from becoming contaminated with copper, otherwise a thin film of copper will form over the surface of the nickel and the chromium will not adhere perfectly. Chromium solutions have poor "throwing power," that is, they will not plate into deep recesses. Therefore, if pieces of irregular shape are to be plated it may be necessary to use anodes of approximately the same shape as the piece to be plated.

Among the recent books and articles published on chromium plating, attention should be called to Electrodeposition of Chromium from Chromic Acid Baths, by H. E. Haring and W. P. Barrows, Bureau of Standards Technologic Paper No. 346 and A Study of Chromium Plating, by R. Schneidewind, University of Michigan Engineering Research Bulletin No. 10.

In the chromium industry there is a real but not critical health-hazard. However, if suitable precautions are taken and sanitary measures and proper ventilation are provided, this hazard should be eliminated. The chief place of attack on the body is on the inside of the nose.

[3] See University of Michigan Engineering Research Bulletin No. 10.

Die Materials and Rare Metals

(Proposed S.A.E. General Information)

Similarly, the Standards Committee is requested to approve the following data on die materials and rare metals for publication in the S.A.E. HANDBOOK as General Information.

Die Materials and Rare Metals

Stellite is a cast alloy containing about 45 per cent of cobalt, 30 per cent of chromium, 15 per cent of tungsten, and small quantities of manganese, silicon, iron and carbon. It has better red-hardness than has high-speed steel but is not so tough. It requires no heat-treatment to develop hardness and cannot be appreciably softened by overheating. It is used extensively in the automotive industry for cutting-tools and dies, but not as a part of automotive equipment. The whole tool may be composed of Stellite, or a Stellite tip may be welded to a less expensive shank. At present the price of stellite is higher than that of high-speed steel.

Cemented Tungsten Carbide is a sintered product usually composed of tungsten carbide, 80 to 97 per cent, and the remainder cobalt. The tungsten carbide usually is made separately in the form of minute particles that are mixed with metallic cobalt powder and pressed to the desired shape. The briquet so formed is then heated in a hydrogen atmosphere to a temperature high enough for the cobalt to sinter the tungsten carbide particles firmly together. The material so produced has a hardness approximating that of sapphire and a toughness suitable for many cutting-tool and die purposes. The red-hardness far surpasses that of any previous cutting-tool material. Cemented tungsten carbide is so expensive at present that small pieces are welded to inexpensive shanks in the manufacture of cutting-tools. Drawing-dies and other dies are usually prepared by mounting small pieces of cemented tungsten carbide in more substantial pieces of other and less expensive material.

Like Stellite, cemented tungsten carbide is used extensively in the manufacture of automotive equipment but is not a part of such equipment. Cemented tungsten carbide requires no heat-treatment to develop the hardness, nor can it be materially softened by over-heating. Its use entails special precautions both as to tool design and operation.

Elkonite is another non-ferrous alloy that is used extensively in the manufacture of automotive equipment but not in the equipment itself. It is a mixture of tungsten and copper with or without some carbon. It is used principally for welding electrodes. Elkonite of several grades is made to fit different requirements. In all of the grades it is desired to maintain high electrical conductivity and high

resistance to wear. It is also necessary to maintain high resistance to upsetting, both at ordinary and elevated temperatures. One grade of Elkonite has a compressive strength of 200,000 lb. per sq. in., a tensile strength of 55,000 lb. per sq. in., and a Brinell hardness-number of 225. Elkonite is a sintered, not cast, product. It is furnished in stock 3/16 to 1 in. square by 8 in. long; in rectangular pieces ⅛ to 1 in. thick with widths up to 3 in. and length up to 8 in.; and in rings up to 8 in. in diameter with width and thickness as required. Elkonite usually is used only as a facing or wearing surface, silver-soldered to a copper electrode. It is usually necessary also to water-cool it.

Miscellaneous

Tungsten is used for make-and-break contacts in automotive ignition systems and incandescent lamps.

Wrought tungsten is used for contact discs. It is essential for best results that the purity be controlled, and it is also desirable to control the grain size. The cross-section of a tungsten rod from which contact discs are cut should show a minimum of 10,000 grains per sq. mm. A longitudinal section will reveal the fibrous structure caused by working below the recrystallization temperature. Contact points are made in five general forms:

(1) Plain rivets
(2) Rivets with shoulders
(3) Screw with head on contact end
(4) Headless screw
(5) Screw with head on thread end.

The thickness of the disc depends somewhat on the diameter. The following standards are suggested:

Diameter, ± 0.002 In.	Thickness, ± 0.002 In.
0.063	0.020
0.093	0.025
0.126	0.030
0.145	0.035
0.151	0.040
0.174	0.045
0.188	0.050
0.251	0.060

The tungsten discs are mounted on the steel screw or rivet by "brazing" with copper or other material, the operation usually being performed in a furnace with a hydrogen atmosphere. Tungsten contacts are used in nearly all battery ignition-systems and in some magnetos.

Filaments used in incandescent lamps described on p. 133, S.A.E. HANDBOOK, 1929 edition, are made of nearly pure tungsten. For most purposes a "non-sag" filament is required. This filament is made by so controlling the process of manufacture that large grains are formed in the filament when first heated to a high temperature in the lamp. Although these large grains make the filament softer when cold, they increase the sag resistance of the filament when heated to the operating temperature. It is not sufficient that the filament be composed of large grains, because the grain boundaries may be so disposed as to allow displacement of adjacent sections by producing the undesirable condition referred to as "off-setting." To avoid off-setting, the grain boundaries should not make right angles with the length of the filament, and to minimize sagging, the grains should be as long as several filament diameters. Not only have these difficult conditions been obtained, but also the same filaments are so tough, either hot or cold, that they withstand high-frequency vibration caused by the engines and any ordinary pump without breaking.

Platinum, alloyed with iridium, is used as a contact material in magnetos where the peak current to be broken exceeds 4 amp. Such magnetos are used especially for aircraft-engine ignition. Because of the high price of the platinum-group metals, the contact surfaces are faced with discs only as thick as necessary for satisfactory operation.

The leading-in wires of incandescent lamps have about the same coefficient of expansion as glass and therefore make a vacuum or gas-tight seal. They consist of a core of iron-nickel alloy containing about 44 per cent of nickel and 56 per cent of iron, coated with copper.

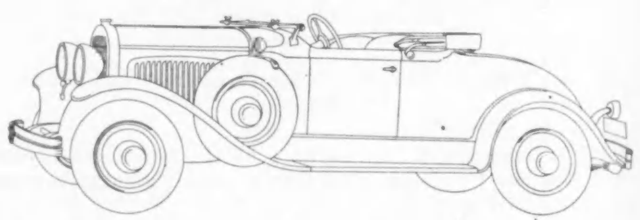
Passenger-Car Division

PERSONNEL

G. L. McCain, *Chairman*	Chrysler Corp.
H. C. Snow, *Vice-Chairman*	Auburn Automobile Co.
S. R. Castor	H. H. Franklin Mfg. Co.
L. A. Chaminade	Studebaker Corp.
W. N. Davis	Cadillac Motor Car Co.
G. A. Delaney	Graham-Paige Motors Corp.
W. T. Fishleigh	Ford Motor Co.
W. H. Graves	Packard Motor Car Co.
J. B. Judkins	J. B. Judkins Co.
E. H. Nollau	E. I. du Pont de Nemours & Co.
Ivan Ornberg	Hupp Motor Car Corp.

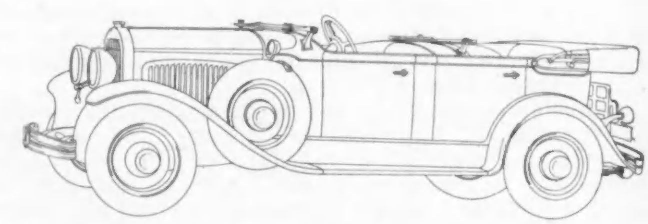
Body-Type Nomenclature

(*Proposed Revision of S.A.E. Standard*)



Roadster.—An open-type body having one cross seat. A compartment in the rear deck accommodates business equipment or luggage. The top is of weatherproof fabric and may be folded. Equipment includes removable side-curtains and provision is usually made for folding the windshield.

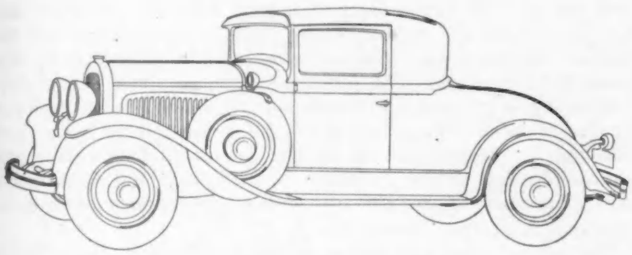
Sport Roadster.—The rear deck is provided with a rumble seat accommodating additional passengers. Equipment frequently includes golf locker in the rear deck. In other respects this type is similar to the Roadster.



Phaeton.—An open-type body with two cross seats, usually accommodating five passengers. Folding-type windshield and folding weatherproof fabric top with removable side-curtains are usual equipment.

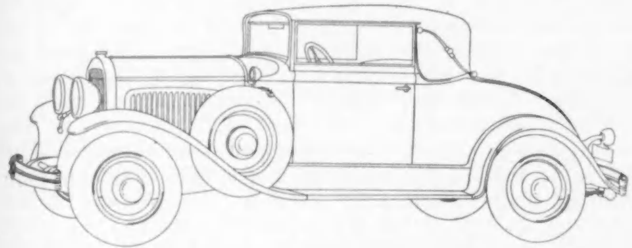
Sport and Imperial Phaeton.—Similar to the Phaeton in general type with various refinements or extra equipment. Wire wheels, trunk rack and ultra-modish finish are common attributes of this type. The Imperial type is accepted to indicate a tonneau windshield.

Touring Car.—Generally longer bodies than the Phaeton, permitting the use of auxiliary seats in the tonneau, for the accommodation of additional passengers. In other respects similar to the Phaeton.

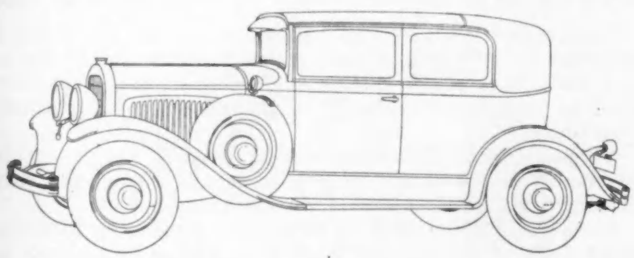


Coupe.—An enclosed single-compartment body. Passenger capacity varies with arrangement of seats or the length of wheelbase. Two doors are provided; back panels and top are permanent and the rear deck accommodates a luggage compartment. Small coupes have a single cross seat accommodating two or three passengers, while the larger coupes frequently provide a staggered seating arrangement which, with an auxiliary seat beside the driver, may accommodate as many as five passengers. The larger types are also generally provided with quarter windows.

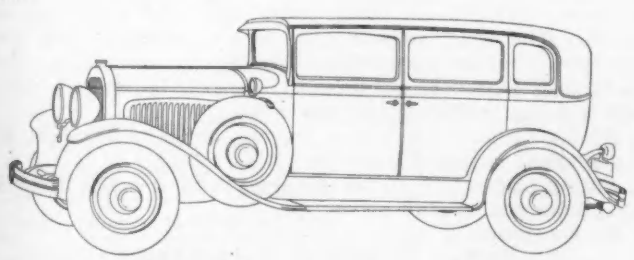
Sport Coupe.—The rear deck is provided with a rumble seat accommodating additional passengers. Equipment frequently includes golf locker in the rear deck. In other respects this type is similar to the Coupe.



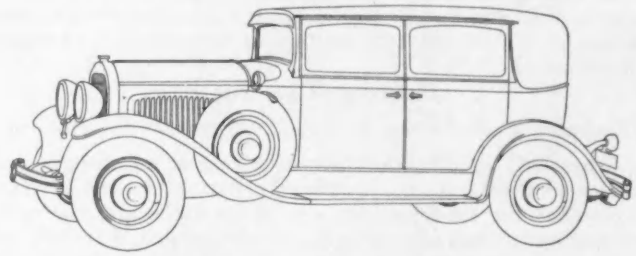
Cabriolet.—Similar to the Sport Coupe with provision for converting to an open-type. The rumble seat and fender wells are usual but not restrictive features of this type.



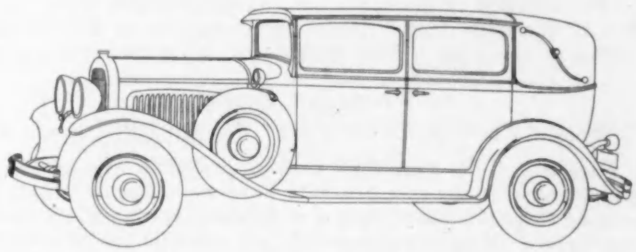
Coach.—An enclosed two-door type body, with permanent back panels and top. A full-width cross seat in the tonneau accommodates three passengers. Two separate seats in the front accommodate the driver and an additional passenger, and by folding down, allow unobstructed exit or entrance to rear-seat passengers. Fender wells and trunk racks are frequently provided but are not inherent features of this type.



Sedan.—An enclosed four-door type body with permanent back panels and top. A full-width cross seat in front and in the tonneau. Passenger capacity from five to seven according to wheelbase. Auxiliary folding-seats in tonneau for accommodation of the extra passengers in the larger types. Usually provided with quarter windows in the rear-quarter deck. Trunk rack and fender wells optional features.

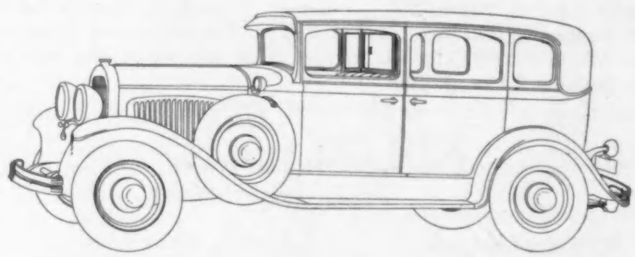


Closed Coupled Sedan.—Similar to the Sedan but with reduced body-length, resulting in closer coupling of the front and rear seats. Passenger capacity generally limited to five. Quarter windows are omitted in this type.



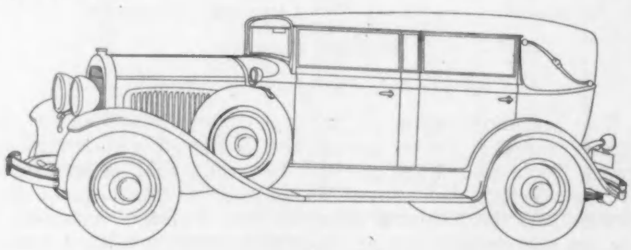
Landau.—A closed-type body with provision for opening or folding the rear quarter. This precludes the use of quarter windows. Landau irons are an inherent but not a distinguishing feature.

Landaulet Sedan.—Similar to the Landau Sedan in appearance but made with a stationary rear-quarter. Landau irons are mounted on the rear quarter but are non-operative.



Imperial Sedan.—A drop or sliding glass partition between the driver's compartment and the tonneau is the distinguishing feature between this type and the Sedan, which it resembles in all other essential respects.

Town Car.—Same as Imperial Sedan with or without rear-quarter windows and without roof over front compartment.



Convertible Sedan.—Similar to Sedan type with provisions for converting to an open-type car. Both the all-weather feature and the top are convertible.

Footman Loops

(Proposed Cancellation of S.A.E. Recommended Practice)

In view of the fact that this item has disappeared from general use in motor-car construction, the Division feels that the specifications are no longer of value and recommends that the S.A.E. Recommended Practice on Footman Loops, p. 360 of the 1929 edition of the S.A.E. HANDBOOK, be cancelled.

Passenger-Car Doors

(Proposed Cancellation of S.A.E. Recommended Practice)

Since this specification covers a matter of design that varies with different types of body construction, the Division feels that the specifications are of no value and therefore recommends that the S.A.E. Recommended Practice on Passenger-Car Doors, p. 360, 1929 edition of the S.A.E. HANDBOOK, be cancelled.

Top-Irons

(Proposed Revision of S.A.E. Recommended Practice)

The Division recommends the changing of the title of the S.A.E. Recommended Practice on Top-Irons, p. 361 of the 1929 edition of the S.A.E. HANDBOOK to Top-Bow Supports.

Wiring for Beads

(Proposed Cancellation of S.A.E. Recommended Practice)

Because of the fact that various practices are in effect and because these specifications have no influence whatsoever on the question of design of fenders, aprons and splash guards, the Division recommends the cancellation of S.A.E. Recommended Practice on Wiring for Beads, p. 361 of the 1929 edition of the S.A.E.HANDBOOK.

Wood Screws

(Proposed Cancellation of S.A.E. Recommended Practice)

The consensus of opinion of the Passenger-Car Division is that the body industry would better be served by a table of wood-screw sizes and dimensions rather than by this specification which fills no need. Therefore, the Division recommends that the present S.A.E. Recommended Practice on Wood Screws, p. 361 of the 1929 edition of the S.A.E. HANDBOOK be cancelled.

Production Division

PERSONNEL

F. W. Stein, *Chairman*	Fairbanks, Morse & Co.
A. R. Fors, *Vice-Chairman*	Continental Motors Corp.
David Ayr	Pratt & Whitney Co. Division, Niles-Bement-Pond Co.
F. P. Barnes	Hupp Motor Car Corp.
F. C. Kroeger	Delco-Remy Corp.
R. R. Lundy	Studebaker Corp.
W. P. Michell	International Motor Co.
D. W. Ovaitt	Buick Motor Co.
L. L. Roberts	Packard Motor Car Co.
E. N. Sawyer	Cleveland Tractor Co.

Milling Cutters

(Proposed American Standard)

The standardization of milling cutters was one of the earlier subjects taken up by the Sectional Committee on Small Tools and Machine-Tool Elements that is sponsored by the Society, the American Society of Mechanical Engineers and the National Machine Tool Builders Association for the standardization of machine tools and related equipment. A careful survey was made of various classes of milling cutters in common use as the basis for the proposed standard, the work being assigned to Technical Committee No. 5 of the Sectional Committee. The supplemental report on milling-cutter nomenclature was also drafted and both preliminary reports were widely circulated by the sponsor organizations for criticism and suggestions by the manufacturing and user industries, these data being duly considered in preparing the report that is now submitted. At the time of this printing of the report, it is being balloted on for final approval by the Sectional Committee and for submission to the sponsors by the Sectional Committee and a number of ballots approving the report have been received from the members.

The report was submitted to the Production Division of the Society's Standards Committee for consideration and recommendation as to approval by the Society as a sponsor in accordance with regular S.A.E. Standards procedure. The Production Division, therefore, now submits the report of the Sectional Committee for approval and for final submission to the American Standards Association for its approval and adoption as American Standard. The report is not submitted at this time for adoption in whole or in part as an S.A.E. Specification.

The recommendation of the Division at the time of printing was subject to confirmation by ballot of the Division members inasmuch as a quorum was not present at the meeting of the Division in December. At this meeting the members of the Division felt that the tolerance of 0.030 in. on the outside diameter of the plain cutters is excessive and also that definite numbers of teeth for fine-tooth cutters and coarse-tooth cutters should be included in the report but that neither of these points are of so great importance as to warrant delaying approval of the report at this time. It was accordingly recommended that when the Sectional Committee on Small Tools and Machine-Tool Elements, through which the report issued, next considers revisions of the report, it change the outside-diameter tolerance for plain cutters from 0.030 to 0.010 in. and include definite numbers of teeth for coarse-tooth and fine-tooth cutters in accordance with adequate data secured from large milling-cutter users.

PART I—NOMENCLATURE

Classification of Milling Cutters, Based on Relief of Teeth

Profile Cutters.—Milling cutters on which the relief is obtained by grinding a narrow land, commonly known as clearance, back of the cutting edge; that is, they are sharpened by grinding the tooth on the periphery of the cutting edges.

Shaped Profile-Cutters.—Milling cutters made to be sharpened in the same manner as profile cutters but with cutting edges of irregular or curved shape.

Formed Cutters.—Milling cutters on which the eccentric relief back of the cutting edge is of the same contour as the cutting edge. These cutters are sharpened by grinding the *face* of the tooth. So long as the face is maintained in its original plane with respect to axis of rotation, the tooth contour will remain unchanged. Formed cutters are usually of curved or irregular outline.

Classification of Milling Cutters, Based on Method of Mounting

Arbor Cutters.—A cutter with hole for mounting on arbor. The most common type have a straight hole and keyway in same. Sometimes keyway is across one end as in shell end mills. Frequently, the hole is tapered. Cutters are also made with threaded hole.

Shank Cutters.—A cutter having either a straight or taper shank integral with the cutter.

Facing Cutter.—A cutter designed to be attached directly to spindle end or stub arbor.

Hand of Rotation of Milling Cutters

The hand of rotation of any cutter may be determined by looking at the cutter end of spindle. If the cutter

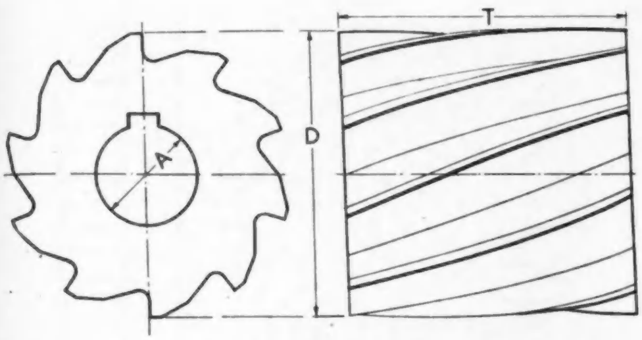


Table No. 1. Plain Milling Cutters (Heavy Duty)

Diameter of Cutter D			Width of Face T			Diameter of Hole A		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2 ½	2.515	2.485	2	2.010	2.000	1	1.001	1.000
2 ½	2.515	2.485	2 ½	2.520	2.500	1	1.001	1.000
2 ½	2.515	2.485	3	3.020	3.000	1	1.001	1.000
2 ½	2.515	2.485	4	4.020	4.000	1	1.001	1.000
3	3.015	2.985	2	2.010	2.000	1 ¼	1.251	1.250
3	3.015	2.985	2 ½	2.520	2.500	1 ¼	1.251	1.250
3	3.015	2.985	3	3.020	3.000	1 ¼	1.251	1.250
3	3.015	2.985	4	4.020	4.000	1 ¼	1.251	1.250
3	3.015	2.985	6	6.020	6.000	1 ¼	1.251	1.250
4	4.015	3.985	2	2.010	2.000	1 ½	1.501	1.500
4	4.015	3.985	3	3.020	3.000	1 ½	1.501	1.500
4	4.015	3.985	4	4.020	4.000	1 ½	1.501	1.500
4	4.015	3.985	6	6.020	6.000	1 ½	1.501	1.500
4 ½	4.515	4.485	6	6.020	6.000	2	2.001	2.000
4 ½	4.515	4.485	12	12.020	12.000	2	2.001	2.000

All dimensions given in inches.
Heavy Duty Cutters have helical teeth.
Hand of spiral optional with cutter manufacturer.
Construction optional with cutter manufacturer.

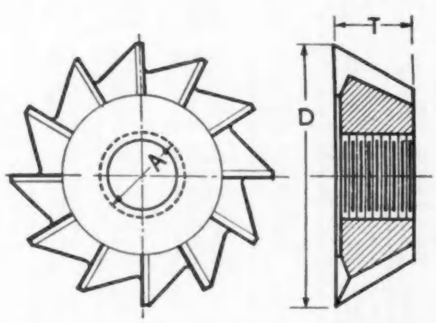


Table No. 8. Single Angle Milling Cutters with Threaded Hole

Diameter of Cutter D			Thickness T			Hole A	
Nom.	Max.	Min.	Nom.	Max.	Min.	Dia.	No. Threads per in.
1 ¼	1 265	1 235	7/16	.452	.422	3/8	24
1 5/8	1.640	1.610	9/16	.577	.547	½	20

All dimensions given in inches.
These cutters will be furnished either right or left hand and with either right or left hand thread.
They have an included angle of 60 degrees.
Tolerance for angle, plus or minus 10 minutes.

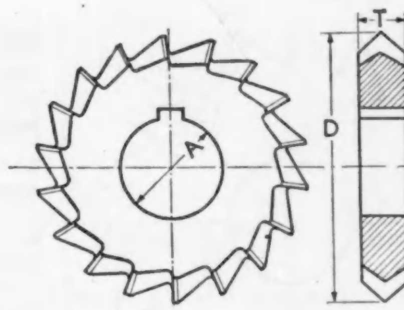


Table No. 9. Double Angle Milling Cutters

Diameter of Cutter D			Thickness T			Diameter of Hole A		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2 ¾	2.765	2.735	½	515	485	1	1 001	1 000

All dimensions given in inches.
Double angle cutters will be furnished with an included angle of either 45, 60 or 90 degrees.
Tolerance for angle, plus or minus 10 minutes.

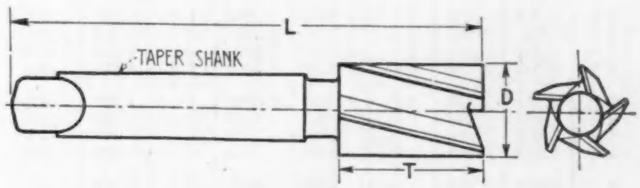


Table No. 10. End Mills—Brown & Sharpe Taper Shank

Diameter of Cutter D			Number of Taper Shank	Length of Cut T	Length Overall L
Nom.	Max.	Min.			
¼	.2650	2500	5	5/8	2 13/16
5/16	.3275	3125	5	11/16	2 7/8
3/8	.3900	.3750	5	¾	2 15/16
7/16	4525	.4375	5	7/8	3 1/16
½	.5150	5000	5	15/16	3 1/8
½	5150	5000	7	15/16	4 15/16
9/16	.5775	5625	7	1	5
5/8	6400	6250	7	1 1/8	5 1/8
¾	7650	7500	7	1 ¼	5 ¼
7/8	8900	8750	7	1 7/16	5 7/16
1	1.0150	1.0000	7	1 5/8	5 5/8
1 1/8	1.1400	1.1250	9	1 ¾	7
1 ¼	1.2650	1.2500	9	2	7 ¼
1 ½	1 5150	1.5000	9	2 ¼	7 ½
1 ¾	1.7650	1.7500	9	2 ½	7 ¾
2	2.0150	2 0000	9	2 ¾	8

All dimensions in inches.
Tolerance for length of cut and length overall, plus or minus 1/32 inch.
End Mills with No. 5 B. & S. taper shank are furnished without tang.
Hand of spiral and construction of end mill is optional with cutter manufacturer.

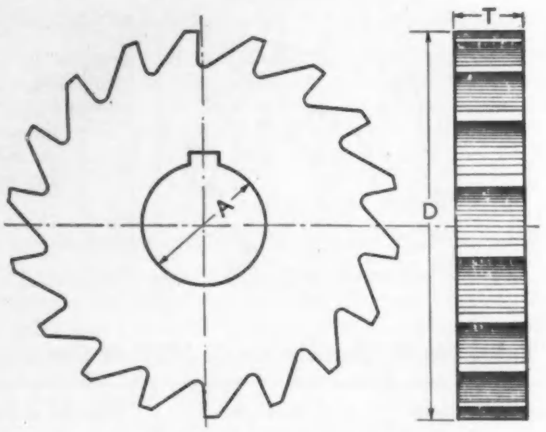


Table No. 2. Plain Milling Cutters (Light Duty)

Diameter of Cutter D			Width of Face T			Diameter of Hole A		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2 1/4	2.265	2.235	1/2	.501	.499	7/8	876	875
2 1/4	2.265	2.235	1	1.0010	.9990	7/8	.876	.875
2 1/2	2.515	2.485	3/16	.1885	.1865	1	1.001	1.000
2 1/2	2.515	2.485	1/4	.2510	.2490	1	1.001	1.000
2 1/2	2.515	2.485	5/16	.3135	3115	1	1.001	1.000
2 1/2	2.515	2.485	3/8	.3760	.3740	1	1.001	1 000
2 1/2	2.515	2.485	7/16	.4385	4365	1	1 001	1.000
2 1/2	2.515	2.485	1/2	.5010	.4990	1	1.001	1.000
2 1/2	2.515	2.485	5/8	.6260	6240	1	1.001	1.000
2 1/2	2.515	2.485	3/4	.7510	.7490	1	1 001	1 000
2 1/2	2.515	2.485	1	1.0010	9990	1	1 001	1 000
2 1/2	2.515	2.485	1 1/2	1.5100	1.5000	1	1.001	1.000
2 1/2	2.515	2.485	2	2.0100	2.0000	1	1 001	1.000
2 1/2	2.515	2.485	2 1/2	2.5200	2 5000	1	1.001	1 000
2 1/2	2.515	2.485	3	3.0200	3 0000	1	1.001	1.000
3	3.015	2.985	3/16	.1885	1865	1	1.001	1.000
3	3.015	2.985	1/4	.2510	2490	1	1.001	1.000
3	3.015	2.985	5/16	.3135	.3115	1	1.001	1.000
3	3.015	2.985	3/8	.3760	.3740	1	1 001	1 000
3	3.015	2.985	3/8	.3760	.3740	1 1/4	1.251	1.250
3	3.015	2.985	7/16	.4385	.4365	1 1/4	1.251	1.250
3	3.015	2.985	1/2	.5010	4990	1 1/4	1 251	1.250
3	3.015	2.985	5/8	.6260	6240	1 1/4	1.251	1 250
3	3.015	2.985	3/4	.7510	.7490	1 1/4	1.251	1.250
3	3.015	2.985	7/8	.8760	.8740	1 1/4	1.251	1.250
3	3.015	2.985	1	1.0010	9990	1 1/4	1.251	1.250
3	3.015	2.985	1 1/4	1.2600	1 2500	1 1/4	1.251	1.250
3	3.015	2.985	1 1/2	1.5100	1 5000	1 1/4	1.251	1.250
3	3.015	2.985	2	2.0100	2.0000	1 1/4	1.251	1.250
3	3.015	2.985	3	3.0200	3 0000	1 1/4	1.251	1.250
3	3.015	2.985	4	4.0200	4.0000	1 1/4	1.251	1.250
3	3.015	2.985	6	6.0200	6.0000	1 1/4	1.251	1.250
4	4.015	3.985	1/4	.2510	.2490	1	1.001	1.000
4	4.015	3.985	1/4	2510	2490	1 1/4	1 251	1.250
4	4.015	3.985	5/16	.3135	.3115	1	1.001	1.000
4	4.015	3.985	5/16	.3135	.3115	1 1/4	1.251	1.250
4	4.015	3.985	3/8	.3760	.3740	1	1.001	1.000
4	4.015	3.985	3/8	.3760	.3740	1 1/4	1.251	1.250
4	4.015	3.985	1/2	5010	4990	1 1/4	1.251	1.250
4	4.015	3.985	5/8	.6260	.6240	1 1/4	1.251	1.250
4	4.015	3.985	3/4	.7510	.7490	1 1/4	1.251	1.250
4	4.015	3.985	1	1.0010	9990	1 1/4	1.251	1.250
4	4.015	3.985	1 1/2	1.5100	1.5000	1 1/4	1.251	1.250
4	4.015	3.985	2	2.0100	2.0000	1 1/4	1.251	1.250
4	4.015	3.985	3	3.0200	3.0000	1 1/4	1.251	1.250
4	4.015	3.985	4	4.0200	4.0000	1 1/4	1.251	1.250
4	4.015	3.985	6	6.0200	6.0000	1 1/2	1.501	1.500

All dimensions given in inches.
Cutters of less than 3/4-inch face have straight teeth.
Cutters of 3/4-inch face and over have helical teeth.
Hand of spiral optional with cutter manufacturer.
Construction optional with cutter manufacturer.

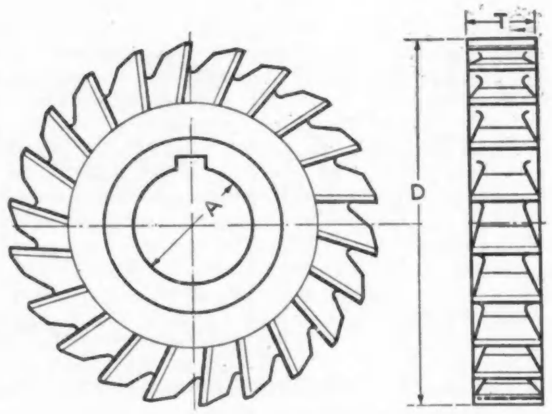


Table No. 3. Side Milling Cutters (Straddle Mills)

Diameter of Cutter D			Width of Face T			Diameter of Hole A		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2	2 015	1 985	3/16	1895	.1865	1/2	501	.500
2	2 015	1 985	3/16	1895	1865	5/8	.626	.625
2	2 015	1.985	1/4	2520	2490	1/2	.501	.500
2	2 015	1.985	1/4	2520	2490	5/8	626	.625
2	2 015	1.985	3/8	.3770	.3740	1/2	.501	.500
2	2 015	1.985	3/8	.3770	.3740	5/8	.626	.625
2 1/2	2 515	2 485	1/4	2520	2490	7/8	.877	.875
2 1/2	2.515	2.485	5/16	3145	.3115	7/8	.877	.875
2 1/2	2 515	2.485	3/8	3770	3740	7/8	.877	.875
2 1/2	2.515	2.485	1/2	5020	4990	7/8	.877	.875
3	3 015	2 985	1/4	2520	.2490	1	1.001	1.000
3	3 015	2 985	5/16	3145	.3115	1	1.001	1.000
3	3 015	2.985	3/8	.3770	.3740	1	1.001	1.000
3	3 015	2.985	7/16	4395	.4365	1	1.001	1.000
3	3 015	2.985	1/2	5020	.4990	1	1 001	1.000
4	4 015	3.985	1/4	.2520	.2490	1	1.001	1.000
4	4.015	3.985	3/8	.3770	.3740	1	1.001	1.000
4	4.015	3.985	1/2	.5020	.4990	1	1.001	1.000
4	4.015	3 985	1/2	.5020	.4990	1 1/4	1.251	1.250
4	4.015	3 985	5/8	.6270	.6240	1	1.001	1.000
4	4 015	3 985	5/8	.6270	.6240	1 1/4	1.251	1.250
4	4 015	3.985	3/4	.7520	.7490	1	1.001	1.000
4	4 015	3 985	3/4	.7520	.7490	1 1/4	1.251	1.250
4	4.015	3.985	7/8	.8770	.8740	1	1.001	1.000
4	4 015	3.985	7/8	.8770	.8740	1 1/4	1.251	1.250
5	5 015	4.985	1/2	5020	.4990	1	1.001	1.000
5	5 015	4.985	1/2	.5020	.4990	1 1/4	1.251	1.250
5	5.015	4.985	5/8	.6270	.6240	1	1.001	1.000
5	5 015	4.985	5/8	.6270	.6240	1 1/4	1.251	1.250
5	5 015	4.985	3/4	.7520	.7490	1	1.001	1.000
5	5.015	4.985	3/4	7520	.7490	1 1/4	1.251	1.250
5	5.015	4.985	1	1.0020	.9990	1 1/4	1.251	1.250
6	6.015	5.985	1/2	.5020	.4990	1	1.001	1.000
6	6.015	5.985	1/2	.5020	4990	1 1/4	1.251	1.250
6	6.015	5.985	5/8	.6270	.6240	1 1/4	1.251	1.250
6	6.015	5.985	3/4	.7520	.7490	1	1.001	1.000
6	6.015	5.985	3/4	7520	7490	1 1/4	1.251	1.250
6	6.015	5.985	1	1.0020	.9990	1 1/4	1.251	1.250
7	7 015	6.985	3/4	.7520	.7490	1 1/4	1.251	1.250
7	7.015	6.985	1	1.0020	.9990	1 1/4	1.251	1.250
8	8.015	7 985	3/4	7520	7490	1 1/4	1.251	1.250
8	8.015	7.985	1	1.0020	9990	1 1/4	1.251	1.250
8	8.015	7.985	1	1 0020	9990	1 1/2	1.501	1.500

All dimensions given in inches.
Construction optional with cutter manufacturer.

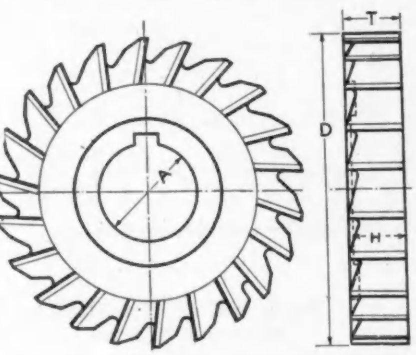


Table No. 4 Half Side Milling Cutter

Diameter of Cutter D			Width of Face T			Diameter of Hole A			Thickness of Hub H		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
4	4.015	3.985	½	.515	.500	1 ¼	1.251	1.250	7/16	.440	.435
4	4.015	3.985	¾	.765	.750	1 ¼	1.251	1.250	9/16	.564	.561
5	5.015	4.985	¾	.765	.750	1 ¼	1.251	1.250	⅝	.627	.624
6	6.015	5.985	¾	.765	.750	1 ¼	1.251	1.250	⅝	.627	.624
6	6.015	5.985	1	1.015	1.000	1 ½	1.501	1.500	⅞	.877	.874
7	7.015	6.985	¾	.765	.750	1 ½	1.501	1.500	⅝	.627	.624
7	7.015	6.985	1	1.015	1.000	1 ½	1.501	1.500	⅞	.877	.874
8	8.015	7.985	¾	.765	.750	1 ½	1.501	1.500	⅝	627	.624
8	8.015	7.985	1	1.015	1.000	1 ½	1.501	1.500	⅞	.877	.874
8	8.015	7.985	1 ¼	1.265	1.250	1 ½	1.501	1.500	1 ⅛	1.127	1.124

All dimensions in inches.
Construction optional with cutter manufacturer.

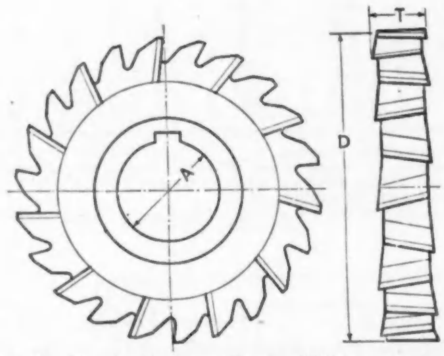


Table No. 5. Stagger Tooth Milling Cutters

Diameter of Cutter D			Width of Face T			Diameter of Hole A		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2 ½	2.515	2.485	¼	.2500	.2495	⅞	.876	.875
2 ½	2.515	2.485	5/16	.3125	.3120	⅞	.876	.875
2 ½	2.515	2.485	⅜	.3750	.3745	⅞	.876	.875
2 ½	2.515	2.485	½	.5000	.4995	⅞	.876	.875
3	3.015	2.985	3/16	.1875	.1870	1	1.001	1.000
3	3.015	2.985	¼	.2500	.2495	1	1.001	1.000
3	3.015	2.985	5/16	.3125	.3120	1	1.001	1.000
3	3.015	2.985	⅜	.3750	.3745	1	1.001	1.000
3	3.015	2.985	½	.5000	.4995	1 ¼	1.251	1.250
3	3.015	2.985	⅝	.6250	.6245	1 ¼	1.251	1.250
3	3.015	2.985	¾	.7500	.7495	1 ¼	1.251	1.250
4	4.015	3.985	¼	.2500	.2495	1 ¼	1.251	1.250
4	4.015	3.985	5/16	.3125	.3120	1 ¼	1.251	1.250
4	4.015	3.985	⅜	.3750	.3745	1 ¼	1.251	1.250
4	4.015	3.985	7/16	.4375	.4370	1 ¼	1.251	1.250
4	4.015	3.985	½	.5000	.4995	1 ¼	1.251	1.250
4	4.015	3.985	⅝	.6250	.6245	1 ¼	1.251	1.250
4	4.015	3.985	¾	.7500	.7495	1 ¼	1.251	1.250
4	4.015	3.985	⅞	.8750	.8740	1 ¼	1.251	1.250
5	5.015	4.985	½	.5000	.4995	1 ¼	1.251	1.250
5	5.015	4.985	⅝	.6250	.6245	1 ¼	1.251	1.250
5	5.015	4.985	¾	.7500	.7495	1 ¼	1.251	1.250
6	6.015	5.985	⅜	.3750	.3745	1 ¼	1.251	1.250
6	6.015	5.985	½	.5000	.4995	1 ¼	1.251	1.250
6	6.015	5.985	⅝	.6250	.6245	1 ¼	1.251	1.250
6	6.015	5.985	¾	.7500	.7495	1 ¼	1.251	1.250
6	6.015	5.985	⅞	.8750	.8740	1 ¼	1.251	1.250
6	6.015	5.985	1	1.0000	.9990	1 ¼	1.251	1.250
8	8.015	7.985	⅜	.3750	.3745	1 ½	1.501	1.250
8	8.015	7.985	½	.5000	.4995	1 ½	1.501	1.500
8	8.015	7.985	⅝	.6250	.6245	1 ½	1.501	1.500
8	8.015	7.985	¾	.7500	.7495	1 ½	1.501	1.500
8	8.015	7.985	1	1.0000	.9990	1 ½	1.501	1.500

All dimensions in inches.
Side teeth are not cutting teeth.
Construction optional with cutter manufacturer.

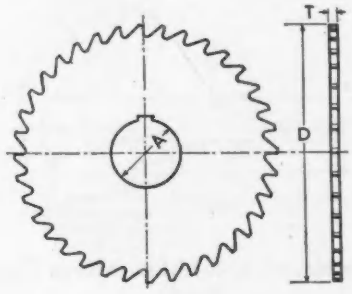


Table No. 6. Metal Slitting Saws

Diameter of Cutter D			Thickness T			Diameter of Hole A		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2 ½	2.515	2.485	1/32	0322	.0302	⅞	.876	875
2 ½	2.515	2.485	3/64	.0478	.0458	⅞	.876	.875
2 ½	2.515	2.485	1/16	.0635	.0615	⅞	.876	.875
2 ½	2.515	2.485	3/32	.0947	.0927	⅞	.876	.875
2 ½	2.515	2.485	⅛	.1260	.1240	⅞	.876	.875
3	3.015	2.985	1/32	.0322	.0302	1	1.001	1.000
3	3.015	2.985	3/64	.0478	.0458	1	1.001	1.000
3	3.015	2.985	1/16	.0635	.0615	1	1.001	1.000
3	3.015	2.985	3/32	.0947	.0927	1	1.001	1.000
3	3.015	2.985	⅛	.1260	.1240	1	1.001	1.000
3	3.015	2.985	5/32	.1572	.1552	1	1.001	1.000
4	4.015	3.985	1/32	.0322	.0302	1	1.001	1.000
4	4.015	3.985	3/64	.0478	.0458	1	1.001	1.000
4	4.015	3.985	1/16	.0635	.0615	1	1.001	1.000
4	4.015	3.985	3/32	.0947	.0927	1	1.001	1.000
4	4.015	3.985	⅛	.1260	.1240	1	1.001	1.000
4	4.015	3.985	5/32	.1572	.1552	1	1.001	1.000
4	4.015	3.985	3/16	.1885	.1865	1	1.001	1.000
5	5.015	4.985	1/16	.0635	.0615	1	1.001	1.000
5	5.015	4.985	3/32	.0947	.0927	1	1.001	1.000
5	5.015	4.985	⅛	.1260	.1240	1	1.001	1.000
5	5.015	4.985	⅛	.1260	.1240	1 ¼	1.251	1.250
5	5.015	4.985	5/32	.1572	.1552	1	1.001	1.000
5	5.015	4.985	3/16	.1885	.1865	1	1.001	1.000
6	6.015	5.985	1/16	0635	.0615	1	1.001	1.000
6	6.015	5.985	3/32	.0947	.0927	1	1.001	1.000
6	6.015	5.985	⅛	.1260	.1240	1	1.001	1.000
6	6.015	5.985	⅛	1260	.1240	1 ¼	1.251	1.250
6	6.015	5.985	3/16	.1885	.1865	1	1.001	1.000
6	6.015	5.985	3/16	.1885	.1865	1 ¼	1.251	1.250
7	7.015	6.985	1/16	.0635	.0615	1	1.001	1.000
7	7.015	6.985	⅛	.1260	.1240	1	1.001	1.000
7	7.015	6.985	⅛	.1260	.1240	1 ¼	1.251	1.250
7	7.015	6.985	3/16	.1885	.1865	1 ¼	1.251	1.250
8	8.015	7.985	⅛	.1260	.1240	1	1.001	1.000
8	8.015	7.985	⅛	.1260	.1240	1 ¼	1.251	1.250
8	8.015	7.985	3/16	.1885	.1865	1 ¼	1.251	1.250

All dimensions in inches.
Construction optional with cutter manufacturer.

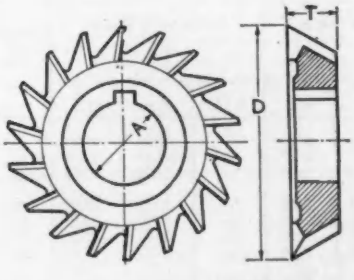


Table No. 7. Single Angle Milling Cutter

Diameter of Cutter D			Thickness T			Diameter of Hole A		
Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2 ½	2.515	2.485	½	.515	.485	⅞	.876	.875
2 ¾	2.765	2.735	½	.515	.485	1	1.001	1.000
3	3.015	2.985	½	.515	.485	1 ¼	1.251	1.250

All dimensions given in inches.
Angular cutters will be furnished either right or left hand with included angle of 45 or 60 degrees.
Tolerance for angle, plus or minus 10 minutes.

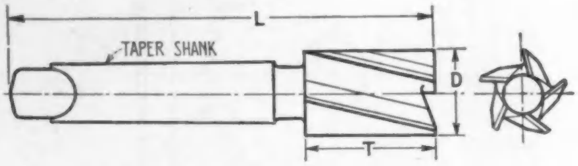


Table No. 11. End Mills—Morse Taper Shank

Diameter of Cutter D			Number of Taper Shank	Length of Cut T	Length Overall L
Nom.	Max.	Min.			
1/4	.2650	.2500	1	5/8	3 1/2
5/16	.3275	.3125	1	11/16	3 9/16
3/8	.3900	.3750	1	3/4	3 5/8
7/16	.4525	.4375	1	7/8	3 3/4
1/2	.5150	.5000	1	15/16	3 13/16
1/2	.5150	.5000	2	15/16	4 7/16
5/8	.6400	.6250	2	1 1/8	4 5/8
3/4	.7650	.7500	2	1 1/4	4 3/4
7/8	.8900	.8750	2	1 7/16	4 15/16
1	1.0150	1.0000	2	1 5/8	5 1/8
3/4	.7650	.7500	3	1 1/4	5 9/16
7/8	.8900	.8750	3	1 7/16	5 3/4
1	1.0150	1.0000	3	1 5/8	5 15/16
1 1/8	1.1400	1.1250	3	1 3/4	6 1/16
1 1/4	1.2650	1.2500	3	2	6 5/16

All dimensions given in inches.
Tolerance for length of cut and length overall, plus or minus 1/32 inch.
Construction optional with cutter manufacturer.
Hand of spiral with reference to hand of cut optional with cutter manufacturer.

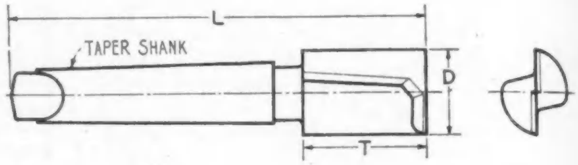


Table No. 13. Two Lipped End Mills Brown & Sharpe Taper Shank

Diameter of Cutter D			Number of Taper Shank	Length of Cut T	Length Overall L
Nom.	Max.	Min.			
1/4	.2510	.2500	5	3/8	2 9/16
5/16	.3135	.3125	5	15/32	2 21/32
1/4	.2510	.2500	7	3/8	4 3/8
5/16	.3135	.3125	7	15/32	4 15/32
3/8	.3760	.3750	7	9/16	4 9/16
7/16	.4385	.4375	7	21/32	4 21/32
1/2	.5010	.5000	7	3/4	4 3/4
9/16	.5635	5625	7	27/32	4 27/32
5/8	.6260	.6250	7	15/16	4 15/16
11/16	.6885	6875	7	1 1/32	5 1/32
3/4	.7510	7500	7	1 1/8	5 1/8
13/16	.8145	.8125	7	1 7/32	5 7/32
7/8	.8780	8750	7	1 5/16	5 5/16
1	1.0020	1.0000	9	1 1/2	6 3/4
1 1/8	1.1280	1.1250	9	1 11/16	6 15/16
1 1/4	1.2530	1.2500	9	1 7/8	7 1/8
1 1/2	1.5020	1.5000	9	2 1/4	7 1/2
1 3/4	1.7520	1.7500	9	2 1/2	7 3/4
2	2.0020	2.0000	9	2 3/4	8

All dimensions in inches.
Tolerance for length of cut and length overall, plus or minus 1/32 inch.
These slotting mills are regularly furnished either right or left hand. Straight or spiral lip optional.
End mills with No. 5 B. & S. taper shank are furnished without tang.

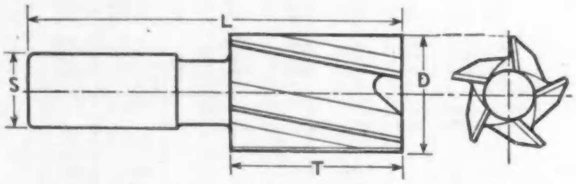


Table No. 12. End Mills—Straight Shank

Diameter of Cutter D			Diameter of Shank S			Length of Cut T	Length Overall L
Nom.	Max.	Min.	Nom.	Max.	Min.		
1/8	.1400	.1250	1/8	.1250	.1245	5/16	1 1/4
5/32	.1713	.1563	5/32	.1563	.1558	5/16	1 1/4
3/16	.2025	.1875	3/16	.1875	.1870	1/2	1 3/8
7/32	.2338	.2188	7/32	.2188	.2183	9/16	1 5/8
1/4	.2650	.2500	1/4	.2500	.2495	5/8	1 11/16
5/16	.3275	.3125	5/16	.3125	.3120	11/16	1 3/4
3/8	.3900	.3750	3/8	.3750	.3745	3/4	1 13/16
7/16	.4525	.4375	7/16	.4375	.4370	7/8	2 3/16
1/2	.5150	.5000	1/2	.5000	.4995	15/16	2 1/4
9/16	.5775	.5625	9/16	.5625	.5620	1	2 5/16
5/8	.6400	.6250	5/8	.6250	.6245	1 1/8	2 1/2
3/4	.7650	.7500	3/4	.7500	.7495	1 1/4	2 5/8

All dimensions given in inches.
Tolerance for length of cut and length overall, plus or minus 1/32 inch.
These end mills are regularly furnished in either right or left hand.
Straight or spiral cut type construction optional with cutter manufacturer.

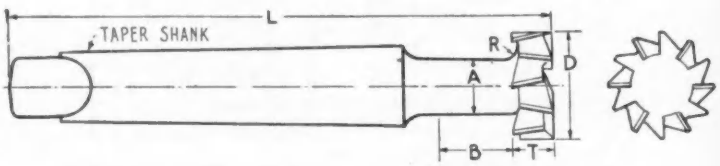


Table 14. T-Slot Cutters—Brown & Sharpe Taper Shank

Nominal Bolt Size	Diameter of Cutter D		Thickness of Cutter T		Diameter of Neck A*		Length of Neck B	Fillet Radius R	Number of Taper Shank	Length Overall L
	Max.	Min.	Max.	Min.	Max.	Min.	Min.	Max.		
1/4	.562	.552	234	.229	.265	260	3/8	.007	5	2 5/8
5/16	.656	.646	265	.260	.328	.323	7/16	007	5	2 23/32
3/8	.781	.771	328	.323	.406	.401	9/16	1/64	7	4 13/16
1/2	.968	.958	.390	.385	.531	.526	11/16	1/64	7	5.
5/8	1.249	1.239	.484	.479	.656	.651	7/8	1/64	7	5 1/4
3/4	1 468	1 458	.625	.620	.781	.776	1 1/16	1/64	9	6 7/8
1	1.843	1.833	.828	.823	1.031	1.026	1 1/4	1/64	9	7 1/4
1 1/4	2.218	2.208	1.093	1.088	1.281	1.276	1 9/16	1/64	9	7 13/16
1 1/2	2 655	2.645	1.343	1 338	1 531	1 526	1 15/16	1/64	10	10 3/8

All dimensions given in inches.
Tolerance: plus or minus 1/64 inch, unless otherwise specified.
T-Slot cutters with No. 5 B. & S. taper shank are furnished without tang.
*Neck diameters as given are to be used with American Standard Tee-Slots.

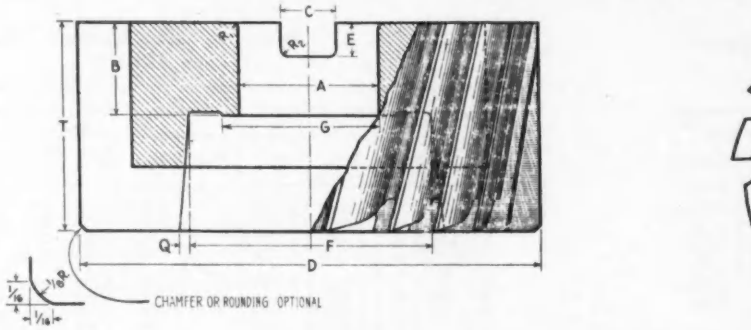


Table No. 15. Shell End Mills

Size or Diam. D	Width of Mill T	Hole: Diameter A Max.	Hole: Diameter A Min.	Hole: Depth B Max.	Hole: Depth B Min.	Driving Slot: Width C Max.	Driving Slot: Width C Min.	Driving Slot: Depth E Max.	Driving Slot: Depth E Min.	Driving Slot: Fillet Radius R	Counterbore: Diam.[1] F	Counterbore: Diam.[1] G	Counterbore: Angular Increase Q deg.
1 1/4	1	.5005	.5000	41/64*	5/8	.262	.258	11/64	5/32	1/64	11/16	5/8	0
1 1/2	1 1/8	.5005	.5000	41/64	5/8	.262	.258	11/64	5/32	1/64	11/16	5/8	0
1 3/4	1 1/4	.7505	.7500	49/64	3/4	.324	.320	13/64	3/16	1/32	15/16	7/8	0
2	1 3/8	.7505	.7500	49/64	3/4	.324	.320	13/64	3/16	1/32	15/16	7/8	0
2 1/4	1 1/2	1.0005	1.0000	49/64	3/4	.387	.383	15/64	7/32	1/32	1 1/4	1 3/16	0
2 1/2	1 5/8	1.0005	1.0000	49/64	3/4	.387	.383	15/64	7/32	1/32	1 3/8	1 3/16	0
2 3/4	1 5/8	1.0005	1.0000	49/64	3/4	.387	.383	15/64	7/32	1/32	1 1/2	1 3/16	.5
3	1 3/4	1.2505	1.2500	49/64	3/4	.512	.508	19/64	9/32	1/32	1 21/32	1 1/2	5
3 1/2	1 7/8	1.2505	1.2500	49/64	3/4	.512	.508	19/64	9/32	1/32	1 11/16	1 1/2	5
4	2 1/4	1.5005	1.5000	1 1/64	1	.637	.633	25/64	3/8	1/16	2 1/32	1 7/8	5
4 1/2	2 1/4	1.5005	1.5000	1 1/64	1	.637	.633	25/64	3/8	1/16	2 1/16	1 7/8	10
5	2 1/4	1.5005	1.5000	1 1/64	1	.637	.633	25/64	3/8	1/16	2 9/16	1 7/8	10
5 1/2	2 1/4	2.0005	2.0000	1 1/64	1	.762	.758	29/64	7/16	1/16	2 13/16	2 1/2	10
6	2 1/4	2.0005	2.0000	1 1/64	1	.762	.758	29/64	7/16	1/16	2 13/16	2 1/2	15

All dimensions in inches.
[1]Tolerances—plus or minus 1/64 inch unless otherwise specified.
Shell End Mills are regularly furnished with spiral teeth in either right or left hand.
Hand of spiral same as hand of cut.

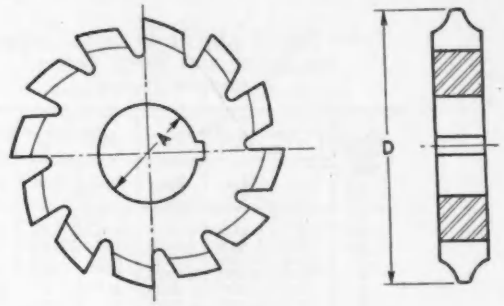


Table No. 16 Involute Gear Cutters[1] Standard Holes

Diametral Pitch	Diameter of Cutter D: Nom.	Diameter of Cutter D: Max.	Diameter of Cutter D: Min.	Diameter of Hole A: Nom.	Diameter of Hole A: Max.	Diameter of Hole A: Min.
*1	8 1/2	8.5625	8.4375	2	2.001	2.000
*1 1/4	7 3/4	7.8125	7.6875	2	2.001	2.000
*1 1/2	7	7.0625	6.9375	1 3/4	1.751	1.750
1 3/4	6 1/2	6.5625	6.4375	1 3/4	1.751	1.750
2	5 3/4	5.8125	5.6875	1 1/2	1.501	1.500
2 1/2	5 3/4	5.8125	5.6875	1 1/2	1.501	1.500
3	4 3/4	4.8125	4.6875	1 1/4	1.251	1.250
4	4 1/4	4.3125	4.1875	1 1/4	1.251	1.250
5	3 3/4	3.8125	3.6875	1 1/4	1.251	1.250
6	3 1/8	3.1875	3.0625	1	1.001	1.000
7	2 7/8	2.9375	2.8125	1	1.001	1.000
8	2 7/8	2.9375	2.8125	1	1.001	1.000
9	2 3/4	2.8125	2.6875	1	1.001	1.000
10	2 3/8	2.4375	2.3125	7/8	.876	.875
11	2 3/8	2.4375	2.3125	7/8	.876	.875
12	2 1/4	2.3125	2.1875	7/8	.876	.875
14	2 1/8	2.1875	2.0625	7/8	.876	.875
16	2 1/8	2.1875	2.0625	7/8	.876	.875
18	2	2.0625	1.9375	7/8	.876	.875
20	2	2.0625	1.9375	7/8	.876	.875
22	2	2.0625	1.9375	7/8	.876	.875
24	1 3/4	1.8125	1.6875	7/8	.876	.875
26	1 3/4	1.8125	1.6875	7/8	.876	.875
28	1 3/4	1.8125	1.6875	7/8	.876	.875
30	1 3/4	1.8125	1.6875	7/8	.876	.875
32	1 3/4	1.8125	1.6875	7/8	.876	.875
36	1 3/4	1.8125	1.6875	7/8	.876	.875
40	1 3/4	1.8125	1.6875	7/8	.876	.875
48	1 3/4	1.8125	1.6875	7/8	.876	.875

All dimensions in inches.
* Cutters not carried in stock, but are made to order.
[1] Note: See statement on bottom of page 35.

Table No. 17 Involute Gear Cutters 1-inch Hole

Diametral Pitch	Diameter of Cutter D: Nom.	Diameter of Cutter D: Max.	Diameter of Cutter D: Min.	Diameter of Hole A: Nom.	Diameter of Hole A: Max.	Diameter of Hole A: Min.
4	3 5/8	3.6875	3.5625	1	1.001	1.000
5	3 3/8	3.4375	3.3125	1	1.001	1.000
10	2 3/4	2.8125	2.6875	1	1.001	1.000
11	2 5/8	2.6875	2.5625	1	1.001	1.000
12	2 5/8	2.6875	2.5625	1	1.001	1.000
14	2 1/2	2.5625	2.4375	1	1.001	1.000
16	2 1/2	2.5625	2.4375	1	1.001	1.000
18	2 3/8	2.4375	2.3125	1	1.001	1.000
20	2 3/8	2.4375	2.3125	1	1.001	1.000
22	2 1/4	2.3125	2.1875	1	1.001	1.000
24	2 1/4	2.3125	2.1875	1	1.001	1.000

All dimensions in inches.

Table No. 18 Involute Gear Cutters $1\frac{1}{4}$-inch Hole

Diametral Pitch	Diameter of Cutter D: Nom.	Diameter of Cutter D: Max.	Diameter of Cutter D: Min.	Diameter of Hole A: Nom.	Diameter of Hole A: Max.	Diameter of Hole A: Min.
6	3 1/2	3.5625	3.4375	1 1/4	1.251	1.250
7	3 3/8	3.4375	3.3125	1 1/4	1.251	1.250
8	3 1/4	3.3125	3.1875	1 1/4	1.251	1.250
9	3 1/8	3.1875	3.0625	1 1/4	1.251	1.250
10	3	3.0625	2.9375	1 1/4	1.251	1.250
12	2 7/8	2.9375	2.8125	1 1/4	1.251	1.250

All dimensions in inches.

Table No. 19 Involute Gear Cutters $1\frac{1}{2}$-Inch Hole

Diametral Pitch	Diameter of Cutter D: Nom.	Diameter of Cutter D: Max.	Diameter of Cutter D: Min.	Diameter of Hole A: Nom.	Diameter of Hole A: Max.	Diameter of Hole A: Min.
3	5 1/4	5.3125	5.1875	1 1/2	1.501	1.500
4	4 1/2	4.5625	5.4375	1 1/2	1.501	1.500
5	4 1/4	4.3125	4.1875	1 1/2	1.501	1.500
6	3 7/8	3.9375	3.8125	1 1/2	1.501	1.500
7	3 5/8	3.6875	3.5625	1 1/2	1.501	1.500
8	3 1/2	3.5625	3.4375	1 1/2	1.501	1.500

All dimensions in inches.

Table No. 20 Involute Gear Cutters $1\frac{3}{4}$-Inch Hole

Diametral Pitch	Diameter of Cutter D: Nom.	Diameter of Cutter D: Max.	Diameter of Cutter D: Min.	Diameter of Hole A: Nom.	Diameter of Hole A: Max.	Diameter of Hole A: Min.
2	6 1/2	6.5625	6.4375	1 3/4	1.751	1.750
2 1/2	6 1/8	6.1875	6.0625	1 3/4	1.751	1.750
3	5 5/8	5.6875	5.5625	1 3/4	1.751	1.750
4	4 3/4	4.8125	4.6875	1 3/4	1.751	1.750
5	4 3/8	4.4375	4.3125	1 3/4	1.751	1.750
6	4 1/4	4.3125	4.1875	1 3/4	1.751	1.750

All dimensions in inches.

Table No. 21 Involute Gear Cutters for Miter and Bevel Gears Standard Holes

Diametral Pitch	Diameter of Cutter D			Diameter of Hole H		
	Nom.	Max.	Min.	Nom.	Max.	Min.
3	4	4.0625	3.9375	1 ¼	1.251	1.250
4	3 ⅝	3.6875	3.5625	1 ¼	1.251	1.250
5	3 ⅜	3.4375	3.3125	1 ¼	1.251	1.250
6	3 ⅛	3.1875	3.0625	1	1.001	1.000
7	2 ⅞	2.9375	2.8125	1	1.001	1.000
8	2 ⅞	2.9375	2.8125	1	1.001	1.000
10	2 ⅜	2.4375	2.3125	⅞	.876	.875
12	2 ¼	2.3125	2.1875	⅞	.876	.875
14	2 ⅛	2.1875	2.0625	⅞	.876	.875
16	2 ⅛	2.1875	2.0625	⅞	.876	.875
20	2	2.0625	1.9375	⅞	.876	.875
24	1 ¾	1.8125	1.6875	⅞	.876	.875

All dimensions in inches.

Involute Gear Cutters

Eight cutters are required for each pitch. These eight cutters are adapted to cut from a pinion of twelve teeth to a rack, and are numbered respectively as follows:

No. 1 will cut gears, 135 teeth to a rack.
No. 2 will cut gears from 55 teeth to 134 teeth.
No. 3 will cut gears from 35 teeth to 54 teeth.
No. 4 will cut gears from 26 teeth to 34 teeth.
No. 5 will cut gears from 21 teeth to 25 teeth.
No. 6 will cut gears from 17 teeth to 20 teeth.
No. 7 will cut gears from 14 teeth to 16 teeth.
No. 8 will cut gears from 12 teeth to 13 teeth.

Cutters listed are for milling gears having 14½° pressure angle. Cutters for other pressure angles will be furnished to order.

Involute gear cutters can be sharpened without changing their outline.

Table No. 22 Involute Gear Cutters Metric Sizes Standard Holes

Module in millimeters	Diameter of Cutter D			Diameter of Hole A			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
½	1 ¾	1.8125	1.6875	⅞	.876	.875	22 m/m.	.8671	.8661
¾	1 ¾	1.8125	1.6875	⅞	.876	.875	22 m/m.	.8671	.8661
1	1 ¾	1.8125	1.6875	⅞	.876	.875	22 m/m.	.8671	.8661
1 ¼	2	2.0625	1.9375	⅞	.876	.875	22 m/m.	.8671	.8661
1 ½	2 ⅛	2.1875	2.0625	⅞	.876	.875	22 m/m.	.8671	.8661
1 ¾	2 ⅛	2.1875	2.0625	⅞	.876	.875	22 m/m.	.8671	.8661
2	2 ¼	2.3125	2.1875	⅞	.876	.875	22 m/m.	.8671	.8661
2 ¼	2 ⅜	2.4375	2.3125	⅞	.876	.875	22 m/m.	.8671	.8661
2 ½	2 ⅜	2.4375	2.3125	⅞	.876	.875	22 m/m.	.8671	.8661
2 ¾	2 ¾	2.8125	2.6875	1	1.001	1.000	27 m/m.	1.064	1.063
3	2 ⅞	2.9375	2.8125	1	1.001	1.000	27 m/m.	1.064	1.063
3 ¼	2 ⅞	2.9375	2.8125	1	1.001	1.000	27 m/m.	1.064	1.063
3 ½	2 ⅞	2.9375	2.8125	1	1.001	1.000	27 m/m.	1.064	1.063
3 ¾	2 ⅞	2.9375	2.8125	1	1.001	1.000	27 m/m.	1.064	1.063
4	3 ⅛	3.1875	3.0625	1	1.001	1.000	27 m/m.	1.064	1.063
4 ¼	3 ⅛	3.1875	3.0625	1	1.001	1.000	27 m/m.	1.064	1.063
4 ½	3 ¾	3.8125	3.6875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
4 ¾	3 ¾	3.8125	3.6875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
5	3 ¾	3.8125	3.6875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
5 ¼	3 ¾	3.8125	3.6875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
5 ½	4	4.0625	3.9375	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
5 ¾	4	4.0625	3.9375	1 ¼	1.251	1.250	22 m/m.	1.2608	1.2598
6	4 ¼	4.3125	4.1875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
7	4 ½	4.5625	4.4375	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
8	4 ¾	4.8125	4.6875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
9	5 ½	5.5625	4.4375	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
10	5 ¾	5.8125	5.6875	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
11	5 ¾	5.8125	5.6875	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
12	5 ¾	5.8125	5.6875	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748

All dimensions given in inches, except as indicated.

Table No. 23 Involute Gear Cutters, Metric Sizes 1-Inch Hole

Module in m/m.	Diameter of Cutter D			Diameter of Hole A			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
¾	2 ¼	2.3125	2.1875	1	1.001	1.000	27 m/m.	1.064	1.063
1	2 ¼	2.3125	2.1875	1	1.001	1.000	27 m/m.	1.064	1.063
1 ¼	2 ⅜	2.4375	2.3125	1	1.001	1.000	27 m/m.	1.064	1.063
1 ½	2 ½	2.5625	2.4375	1	1.001	1.000	27 m/m.	1.064	1.063
1 ¾	2 ½	2.5625	2.4375	1	1.001	1.000	27 m/m.	1.064	1.063
2	2 ⅝	2.6875	2.5625	1	1.001	1.000	27 m/m.	1.064	1.063
2 ¼	2 ⅝	2.6875	2.5625	1	1.001	1.000	27 m/m.	1.064	1.063
2 ½	2 ¾	2.8125	2.6875	1	1.001	1.000	27 m/m.	1.064	1.063
4 ½	3 ¼	3.3125	3.1875	1	1.001	1.000	27 m/m.	1.064	1.063
5	3 ⅜	3.4375	3.3125	1	1.001	1.000	27 m/m.	1.064	1.063
5 ½	3 ½	3.5625	3.4375	1	1.001	1.000	27 m/m.	1.064	1.063
6	3 ⅝	3.6875	3.5625	1	1.001	1.000	27 m/m.	1.064	1.063

All dimensions in inches, except as indicated.

Table No. 24 Involute Gear Cutters, Metric Sizes 1¼-Inch Hole

Module in m/m.	Diameter of Cutter D			Diameter of Hole A			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
1 ¼	2 ¾	2.8125	2.6875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
1 ½	2 ⅞	2.9375	2.8125	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
1 ¾	2 ⅞	2.9375	2.8125	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
2	2 ⅞	2.9375	2.8125	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
2 ¼	2 ⅞	2.9375	2.8125	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
2 ½	3	3.0625	2.9375	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
3	3 ¼	3.3125	3.1875	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
3 ½	3 ⅜	3.4375	3.3125	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598
4	3 ½	3.5625	3.4375	1 ¼	1.251	1.250	32 m/m.	1.2608	1.2598

All dimensions in inches, except as indicated.

Table No. 25 Involute Gear Cutters, Metric Sizes 1½-Inch Hole

Module in m/m.	Diameter of Cutter D			Diameter of Hole A			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.	Nom.	Max.	Min.
2 ½	3 ½	3.5625	3.4375	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
3	3 ½	3.5625	3.4375	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
3 ½	3 ⅝	3.6875	3.5625	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
4	3 ⅞	3.9375	3.8125	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
4 ½	4 ⅛	4.1875	4.0625	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
5	4 ¼	4.3125	4.1875	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
5 ½	4 ⅜	4.4375	4.3125	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
6	4 ½	4.5625	4.4375	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
7	4 ⅞	4.9375	4.8125	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748
8	5 ¼	5.3125	5.1875	1 ½	1.501	1.500	40 m/m.	1.5758	1.5748

All dimensions in inches, except as indicated.

Table No. 26 Involute Gear Cutters, Metric Sizes 1¾-Inch Hole

Module in m/m.	Diameter of Cutter D			Diameter of Hole A			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min	Nom.	Max.	Min.
3	4	4.0625	3.9375	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
3 ½	4 ⅛	4.1875	4.0625	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
4	4 ¼	4.3125	4.1875	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
4 ½	4 ⅜	4.4375	4.3125	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
5	4 ⅜	4.4375	4.3125	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
5 ½	4 ⅝	4.6875	4.5625	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
6	4 ¾	4.8125	4.6875	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
7	5 ¼	5.3125	5.1875	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
8	5 ⅝	5.6875	5.5625	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
9	5 ⅞	5.9375	5.8125	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
10	6 ⅛	6.1875	6.0625	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
11	6 ½	6.5625	6.4375	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716
12	6 ½	6.5625	6.4375	1 ¾	1.751	1.750	45 m/m.	1.7726	1.7716

All dimensions in inches, except as indicated.

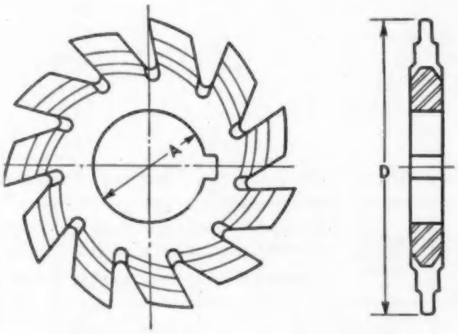


Table No. 27 Stocking Cutters for Involute Gears[1] Standard Holes

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
* 1	8 ½	8.5625	8.4375	2	2.001	2.000
* 1 ¼	7 ¾	7.8125	7.6875	2	2.001	2.000
* 1 ½	7	7.0625	6.9375	1 ¾	1.751	1.750
1 ¾	6 ½	6.5625	6.4375	1 ¾	1.751	1.750
2	5 ¾	5.8125	5.6875	1 ½	1.501	1.500
2 ½	5 ¾	5.8125	5.6875	1 ½	1.501	1.500
3	4 ¾	4.8125	4.6875	1 ¼	1.251	1.250
4	4 ¼	4.3125	4.1875	1 ¼	1.251	1.250
5	3 ¾	3.8125	3.6875	1 ¼	1.251	1.250
6	3 ⅛	3.1875	3.0625	1	1.001	1.000
7	2 ⅞	2.9375	2.8125	1	1.001	1.000
8	2 ⅞	2.9375	2.8125	1	1.001	1.000

All dimensions in inches.
Cutters marked * are not kept in stock, but are made to order.

[1]Note: One cutter is made for each diametral pitch. Cutters listed are for milling gears having 14½ degree pressure angle, but other pressure angles will be furnished on order. Stocking Cutters may be sharpened without changing their form.

Table No. 28 Stocking Cutters for Involute Gears 1-Inch Hole

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
4	3 ⅝	3.6875	3.5625	1	1.001	1.000
5	3 ⅜	3.4375	3.3125	1	1.001	1.000

All dimensions in inches.

Table No. 29 Stocking Cutters for Involute Gears 1¼-Inch Hole

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
6	3 ½	3.5625	3.4375	1 ¼	1.251	1.250
7	3 ⅜	3.4375	3.3125	1 ¼	1.251	1.250
8	3 ¼	3.3125	3.1875	1 ¼	1.251	1.250

All dimensions in inches.

Table No. 30 Stocking Cutters for Involute Gears 1½-Inch Hole

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
3	5 ¼	5.3125	5.1875	1 ½	1.501	1.500
4	4 ½	4.5625	5.4375	1 ½	1.501	1.500
5	4 ¼	4.3125	4.1875	1 ½	1.501	1.500
6	3 ⅞	3.9375	3.8125	1 ½	1.501	1.500

All dimensions in inches.

Table No. 31 Stocking Cutters for Involute Gears 1¾-Inch Hole

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
2	6 ½	6.5625	6.4375	1 ¾	1.751	1.750
2 ½	6 ⅛	6.1875	6.0625	1 ¾	1.751	1.750
3	5 ⅝	5.6875	5.5625	1 ¾	1.751	1.750
4	4 ¾	4.8125	4.6875	1 ¾	1.751	1.750
5	4 ⅜	4.4375	4.3125	1 ¾	1.751	1.750

All dimensions in inches.

Table No. 32 Involute Gear Cutters With Large Diameters 1¼-Inch Hole

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
2	5	5.0625	4.9375	1 ¼	1.251	1.250
3	4 ¾	4.8125	4.6875	1 ¼	1.251	1.250
4	4	4.0625	3.9375	1 ¼	1.251	1.250
5	3 ½	3.5625	3.4375	1 ¼	1.251	1.250
6	3 ½	3.5625	3.4375	1 ¼	1.251	1.250
7	3 ½	3.5625	3.4375	1 ¼	1.251	1.250
8	3 ½	3.5625	3.4375	1 ¼	1.251	1.250
9	3 ½	3.5625	3.4375	1 ¼	1.251	1.250
10	3 ½	3.5625	3.4375	1 ¼	1.251	1.250
12	3 ½	3.5625	3.4375	1 ¼	1.251	1.250

All dimensions in inches.
For Notes, see Sheet (1), Involute Gear Cutters.

Table No. 33 Involute Gear Cutters With Large Diameters 1½-Inch Hole

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
* 1	8 ½	8.5625	8.4375	1 ½	1.501	1.500
* 1 ¼	7 ¾	7.8125	7.6875	1 ½	1.501	1.500
* 1 ½	7 ¼	7.3125	7.1875	1 ½	1.501	1.500
1 ¾	6 ¾	6.8125	6.6875	1 ½	1.501	1.500
2	6 ¼	6.3125	6.1875	1 ½	1.501	1.500
2 ¼	6 ¼	6.3125	6.1875	1 ½	1.501	1.500
2 ½	6 ¼	6.3125	6.1875	1 ½	1.501	1.500
2 ¾	6 ¼	6.3125	6.1875	1 ½	1.501	1.500
3	5 ¼	5.3125	5.1875	1 ½	1.501	1.500
4	5 ¼	5.3125	5.1875	1 ½	1.501	1.500
5	5 ¼	5.3125	5.1875	1 ½	1.501	1.500
6	4 ¼	4.3125	4.1875	1 ½	1.501	1.500
7	4 ¼	4.3125	4.1875	1 ½	1.501	1.500
8	4 ¼	4.3125	4.1875	1 ½	1.501	1.500
10	4 ¼	4.3125	4.1875	1 ½	1.501	1.500
12	4 ¼	4.3125	4.1875	1 ½	1.501	1.500
14	4 ¼	4.3125	4.1875	1 ½	1.501	1.500
16	4 ¼	4.3125	4.1875	1 ½	1.501	1.500

Table No. 34 Involute Gear Cutters With Large Diameters 2-Inch Hole

Diametral Pitch	Diameter of Cutter D			Diameter of Hole A		
	Nom.	Max.	Min.	Nom.	Max.	Min.
* 1	8 ½	8.5625	8.4375	2	2.001	2.000
* 1 ¼	7 ¾	7.8125	7.6875	2	2.001	2.000
* 1 ½	7 ¼	7.3125	7.1875	2	2.001	2.000
1 ¾	6 ¾	6.8125	6.6875	2	2.001	2.000
2	6 ¼	6.3125	6.1875	2	2.001	2.000
2 ¼	6 ¼	6.3125	6.1875	2	2.001	2.000
2 ½	6 ¼	6.3125	6.1875	2	2.001	2.000
2 ¾	6 ¼	6.3125	6.1875	2	2.001	2.000
3	5 ¼	5.3125	5.1875	2	2.001	2.000
4	5 ¼	5.3125	5.1875	2	2.001	2.000
5	5 ¼	5.3125	5.1875	2	2.001	2.000
6	4 ¼	4.3125	4.1875	2	2.001	2.000
7	4 ¼	4.3125	4.1875	2	2.091	2.000
8	4 ¼	4.3125	4.1875	2	2.001	2.000
10	4 ¼	4.3125	4.1875	2	2.001	2.000
12	4 ¼	4.3125	4.1875	2	2.001	2.000
14	4 ¼	4.3125	4.1875	2	2.001	2.000
* 16	4 ¼	4.3125	4.1875	2	2.001	2.000

All dimensions in inches.
Cutters marked * are not kept in stock, but are made to order.

Table No. 37 Cutters for Fluting Taps and Reamers

Cutter Number	Diameter of Reamer Inches	Number of Teeth in Reamer	Diameter of Cutter D Nom.	Diameter of Cutter D Max.	Diameter of Cutter D Min.	Diameter of Hole A Nom.	Diameter of Hole A Max.	Diameter of Hole A Min.
1	1/8- 1/4	6	2	2.0100	1.9900	1	1.0010	1.0000
2	9/32- 3/8	6	2	2.0100	1.9900	1	1.0010	1.0000
3	13/32- 1/2	6	2 1/8	2.1350	2.1150	1	1.0010	1.0000
4	17/32-1 1/8	6-8	2 1/4	2.2600	2.2400	1	1.0010	1.0000
5	1 5/32-1 3/4	8-10	2 3/4	2.7600	2.7400	1 1/4	1.2510	1.2500
6	1 25/32-2	10	3	3.0100	2.9900	1 1/4	1.2510	1.2500
7	2 1/16-2 1/2	10	3 1/8	3.1350	3.1150	1 1/4	1.2510	1.2500
8	2 9/16-3	10	3 3/8	3.3850	3.3650	1 1/4	1.2510	1.2500

All dimensions in inches.
These Cutters have formed teeth and can be sharpened without changing their outline. Cutters having dimensions other than listed are special.

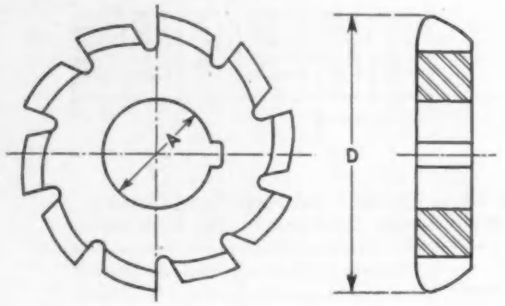


Table No. 35 Cutters for Fluting Reamers

Cutter Number	Diameter of Reamer Inches	Number of Teeth in Reamer	Diameter of Cutter D Nom.	Diameter of Cutter D Max.	Diameter of Cutter D Min.	Diameter of Hole A Nom.	Diameter of Hole A Max.	Diameter of Hole A Min.
1	1/8 - 3/16	6	2 1/4	2.2600	2.2400	1	1.0010	1.0000
2	1/4 - 5/16	6	2 1/4	2.2600	2.2400	1	1.0010	1.0000
3	3/8 - 7/16	6	2 3/8	2.3850	2.3650	1	1.0010	1.0000
4	1/2 -11/16	6-8	2 1/2	2.5100	2.4900	1	1.0010	1.0000
5	3/4 -1	8	2 7/8	2.8850	2.8650	1 1/4	1.2510	1.2500
6	1 1/16-1 1/2	10	3	3.0100	2.9900	1 1/4	1.2510	1.2500
7	1 9/16-2 1/8	12	3 1/8	3.1350	2.1150	1 1/4	1.2510	1.2500
8	2 1/4 -3	14	3 1/4	3.2600	3.2400	1 1/4	1.2510	1.2500

All dimensions in inches.
These Cutters have formed teeth and can be sharpened without changing their outline. Cutters having dimensions other than listed are special.

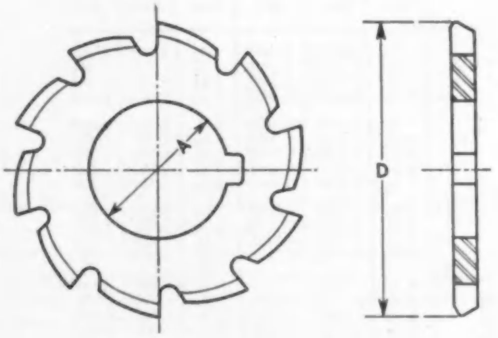


Table No. 36 Cutters for Fluting Taps and Reamers

Cutter Number	Diameter of Tap Inches	Number of Flutes in Tap	Diameter of Cutter D Nom.	Diameter of Cutter D Max.	Diameter of Cutter D Min.	Diameter of Hole A Nom.	Diameter of Hole A Max.	Diameter of Hole A Min.
1	0 - 1/8	4	2	2.0100	1.9900	1	1.0010	1.0000
2	5/32- 1/4	4	2	2.0100	1.9900	1	1.0010	1.0000
3	9/32- 3/8	4	2 1/8	2.1350	2.1150	1	1.0010	1.0000
4	7/16- 5/8	4	2 1/4	2.2600	2.2400	1	1.0010	1.0000
5	11/16- 7/8	4	2 3/4	2.7600	2.7400	1 1/4	1.2510	1 2500
6	15/16-1 1/4	4	3	3.0100	2.9900	1 1/4	1.2510	1.2500
7	1 5/16-1 5/8	4	3 1/8	3.1350	3.1150	1 1/4	1.2510	1 2500
8	1 11/16-2	4	3 3/8	3.3850	3.3650	1 1/4	1 2510	1.2500

All dimensions in inches.
These Cutters have formed teeth and can be sharpened without changing their outline. Cutters having dimensions other than listed are special.

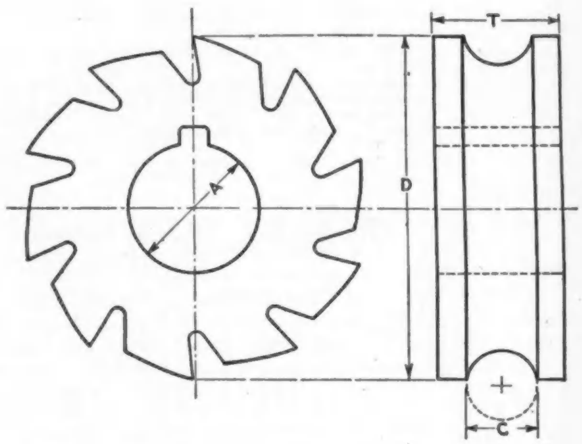


Table No. 38 Concave Cutters

Diameter of Circle C Nominal	Diameter of Circle C Maximum	Diameter of Circle C Minimum	Diameter of Cutter D	Thickness of Cutter T ± .010	Diameter of Hole A Nominal	Diameter of Hole A Maximum	Diameter of Hole A Minimum
1/8	.1280	.1240	2	1/4	7/8	.876	.875
3/16	.1905	.1865	2	3/8	7/8	.876	.875
1/4	.2530	.2490	2	7/16	7/8	.876	.875
5/16	.3155	.3115	2 1/4	9/16	7/8	.876	.875
3/8	.3780	.3740	2 1/4	5/8	7/8	.876	.875
7/16	.4405	.4365	2 1/4	3/4	7/8	.876	.875
1/2	.5040	.4980	2 1/4	13/16	7/8	.876	.875
5/8	.6290	.6230	2 3/4	1	1	1.001	1.000
3/4	.7540	.7480	3	1 3/16	1	1.001	1.000
7/8	.8790	.8730	3 1/4	1 3/8	1	1.001	1.000
1	1.0050	.9980	3 1/4	1 9/16	1	1.001	1.000

All dimensions in inches.
Radial or hooked teeth optional with manufacturer.

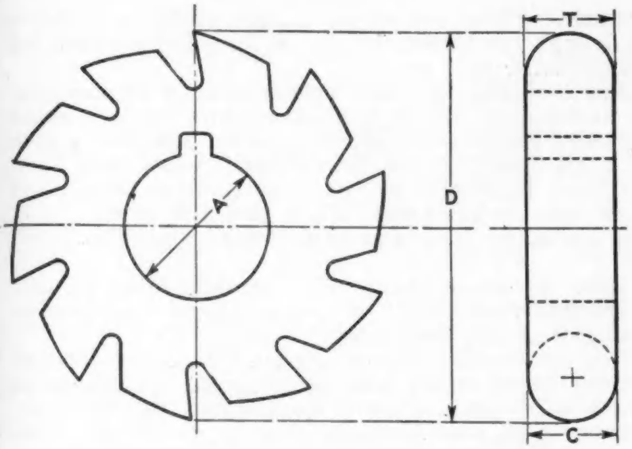


Table No. 39 Convex Cutter

Diameter of Circle C			Diameter of Cutter D	Thickness T Nominal	Diameter of Hole A		
Nominal	Maximum	Minimum			Nominal	Maximum	Minimum
1/8	.1270	.1230	2	1/8	7/8	.876	.875
3/16	.1895	.1855	2	3/16	7/8	876	.875
1/4	.2520	.2480	2	1/4	7/8	876	.875
5/16	.3145	.3105	2 1/4	5/16	7/8	876	.875
3/8	.3770	.3740	2 1/4	3/8	7/8	876	.875
7/16	.4395	.4355	2 1/4	7/16	7/8	876	.875
1/2	.5030	.4970	2 1/4	1/2	7/8	.876	.875
5/8	.6280	.6220	2 3/4	5/8	1	1.001	1.000
3/4	.7530	.7470	3	3/4	1	1.001	1.000
7/8	.8780	.8720	3 1/4	7/8	1	1.001	1.000
1	1.0040	.9960	3 1/4	1	1	1.001	1.000

All dimensions in inches.
Radial or hooked teeth optional with manufacturer.

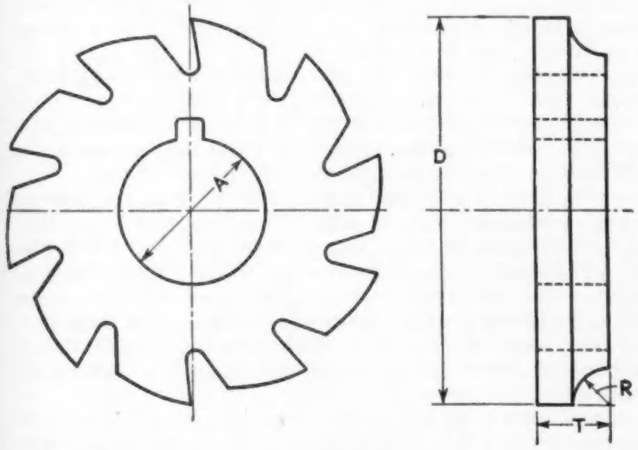


Table No. 40 Corner Rounding Cutters

Radius of Circle R			Diameter of Cutter D	Thickness of Cutter T ± .010	Diameter of Hole A		
Nominal	Maximum	Minimum			Nominal	Maximum	Minimum
1/8	.1265	.1245	2	1/4	7/8	.876	.875
1/4	.2520	.2490	2 1/4	13/32	7/8	.876	.875
3/8	.3770	.3740	3	9/16	1	1.001	1.000
1/2	.5025	.4990	3 1/4	3/4	1	1.001	1.000
5/8	.6275	.6240	3 1/2	15/16	1	1.001	1.000

All dimensions in inches.
Furnished either right or left hand.
Radial or hooked teeth optional with manufacturer.

Table No. 41 Sprocket Wheel Cutters for Roller Chains, American National Standard Tooth Form

Circular Pitch Inches	Diameter of Roll Inches	No. of Teeth in Sprocket	Diameter of Cutter Inches	Width of Cutter Inches	Size of Hole Inches
3/8	200	6	2 3/4	15/32	1
		7-8	2 3/4	15/32	
		9-11	2 3/4	15/32	
		12-17	2 3/4	7/16	
		18-34	2 3/4	7/16	
		35 and over	2 3/4	13/32	
1/2 to 5/8	313	6	3	3/4	1
		7-8	3	3/4	
		9-11	3 1/8	3/4	
		12-17	3 1/8	3/4	
		18-34	3 1/8	23/32	
		35 and over	3 1/8	11/16	
5/8	400	6	3 1/8	3/4	1
		7-8	3 1/8	3/4	
		9-11	3 1/4	3/4	
		12-17	3 1/4	3/4	
		18-34	3 1/4	23/32	
		35 and over	3 1/4	11/16	
3/4	469	6	3 1/4	29/32	1
		7-8	3 1/4	29/32	
		9-11	3 3/8	29/32	
		12-17	3 3/8	7/8	
		18-34	3 3/8	27/32	
		35 and over	3 3/8	13/16	
1	563	6	3 3/4	1 1/4	1 1/4
		7-8	3 7/8	1 1/4	
		9-11	3 7/8	1 3/16	
		12-17	4	1 5/32	
		18-34	4	1 1/8	
		35 and over	4	1 3/32	
1 to 1 1/4	625	6	3 7/8	1 1/2	1 1/4
		7-8	4	1 1/2	
		9-11	4 1/8	1 15/32	
		12-17	4 1/8	1 15/32	
		18-34	4 1/4	1 13/32	
		35 and over	4 1/4	1 11/32	
1 1/4 to 1 1/2	750	6	4 1/4	1 13/16	1 1/4
		7-8	4 3/8	1 13/16	
		9-11	4 1/2	1 25/32	
		12-17	4 1/2	1 3/4	
		18-34	4 5/8	1 11/16	
		35 and over	4 5/8	1 5/8	
1 1/2	875	6	4 3/8	1 13/16	1 1/4
		7-8	4 1/2	1 13/16	
		9-11	4 5/8	1 25/32	
		12-17	4 5/8	1 3/4	
		18-34	4 3/4	1 11/16	
		35 and over	4 3/4	1 5/8	
1 3/4	1.000	6	5	2 3/32	1 1/2
		7-8	5 1/8	2 3/32	
		9-11	5 1/4	2 1/16	
		12-17	5 3/8	2 1/32	
		18-34	5 1/2	1 31/32	
		35 and over	5 1/2	1 7/8	
2	1.125	6	5 3/8	2 13/32	1 1/2
		7-8	5 1/2	2 13/32	
		9-11	5 5/8	2 3/8	
		12-17	5 3/4	2 5/16	
		18-34	5 7/8	2 1/4	
		35 and over	5 7/8	2 5/32	
2 1/2	1 5625	6	6 3/8	3	1 3/4
		7-8	6 5/8	3	
		9-11	6 3/4	2 15/16	
		12-17	6 7/8	2 29/32	
		18-34	7	2 3/4	
		35 and over	7 1/8	2 11/16	
3	1.900	6	7 1/2	3 19/32	2
		7-8	7 3/4	3 19/32	
		9-11	7 7/8	3 17/32	
		12-17	8	3 15/32	
		18-34	8	3 11/32	
		35 and over	8 1/4	3 7/32	

All dimensions in inches.

rotates counter-clockwise, it is right hand; if it rotates clockwise, it is left hand.

Definitions of Various Types of Milling Cutter

Plain Milling-Cutter or Slabbing Mill.—Cutter of plain cylindrical form having teeth on the circumferential surface only. Teeth may be either straight or helical.

Side Milling-Cutter or Straddle Mill.—Cutter of cylindrical form, having teeth on the circumferential surface and also on both sides. The side teeth extend a portion of the distance from the circumference toward the axis. These cutters are frequently used in pairs for milling both ends of work to a given dimension.

Half-Side Milling-Cutter.—Cutters of cylindrical form having teeth on the circumferential surface and teeth on one side only. The side teeth extend a portion of the distance from circumference toward the axis. These cutters are frequently used in pairs for milling both ends of work to a given dimension. Hand of rotation is determined as defined under that heading.

Interlocking Side-Milling Cutter.—Similar in design to a side-milling cutter except made in a *unit* of two interlocking sections for the purpose of milling slots to exact width. Maintained at constant width by use of thin shims or collars between inner hubs.

Staggered-Tooth Milling or Alternate-Tooth Cutter.—Cutter of cylindrical form having *cutting* teeth on the circumferential surface only, the teeth being alternately of opposite helix or angle. Side teeth extending from circumference a short distance toward axis are for chip clearance only but are not ground for cutting purposes. Used for obtaining exact width of slots and is most efficient for milling slots where depth exceeds width.

Metal-Slitting Saw.—Plain milling-cutter with sides relieved or "dished" to afford side clearance, generally made in thickness of 3/16 in. or less, and generally having more teeth for a given diameter than a plain milling-cutter. Used for cutting off work, or milling very narrow slots.

Metal-Slitting Saw with Side Teeth.—Similar to side milling-cutter but 3/16 in. or less in thickness.

Metal-Slitting Saw with Staggered Teeth.—Similar to staggered-tooth milling-cutter but generally 3/8 to 3/16 in. in thickness, used for heavy sawing in steel.

Screw-Slotting Cutter.—A thin cutter made of sheet stock having comparatively fine teeth on its circumferential surface and not ground on the sides. Used only for shallow cuts.

Single-Angle Milling-Cutter.—Cutter having teeth on the conical surface and with or without teeth on one or both of the flat sides. The *included* angle between the conical face and larger flat face designates the cutter as for example 45 or 60 deg. When viewed from the larger flat side, the hand of rotation is determined according to definition previously given for Hand of Rotation.

Double-Angle Milling-Cutter.—Cutter having two intersecting conical surfaces with teeth on both. Angle of teeth may or may not be symmetrical with respect to a plane at right angles to axis. Symmetrical-angle cutters are designated by included angle of tooth. Unsymmetrical-angle cutters are designated by specifying angle of each side with plane of intersection at right angles to axis. In case of unsymmetrical angle when cutter is viewed from the larger flat side, hand of rotation is determined as defined under that heading.

End Mill.—Cutter with teeth on circumferential surface and one end, having integral shank, either straight or taper, for driving. The teeth may be parallel to axis of rotation or helical and either right or left hand. The hand of rotation is determined by viewing end teeth; if counter-clockwise, right hand; if clockwise, left hand. End mill with moderate helix angle is commonly referred to as a spiral end mill.

Two-Lip End or Slotting Mill.—A shank cutter with two cutting teeth on circumferential surface and end teeth cut to center. Flutes are either straight or helical. Cutter can be sunk directly into material to be milled and then fed longitudinally.

Shell End-Mill.—A cutter having teeth on circumferential surface and on one end. The tooth end is recessed to receive nut or screwhead for holding cutter on a stub arbor. Generally driven from keyslot across back face. Teeth may be parallel to axis of rotation, or helical, and either right or left hand. When viewed from end teeth, hand of rotation is determined as defined under that heading.

T-Slot Cutter.—A shank, either straight or taper, cutter designed for milling T-slots having teeth on circumferential surface and both sides.

Woodruff Key-Seat Cutter.—Shank-type, either straight or taper, cutter having teeth generally on circumferential surface only with sides slightly concaved for clearance. Hole type, the style generally used in sizes larger than 2-in. diameter. These cutters are also made with stagger teeth. Both types are used for the specific purpose of milling the semi-cylindrical keyways in shafts for the insertion of Woodruff keys.

Hollow Mill.—A cutter of tubular construction having teeth on one end and internal clearance. The internal clearance is sometimes obtained by plain tapered hole having back taper and sometimes by internal cleared flutes. Generally used for sizing cylindrical stock or machining straight ends of work.

Gear Cutter.—Formed cutter for cutting one space at a time in gears.

Multiple Gear-Cutter.—A single-unit formed-cutter or two or more formed-cutters made to mill two or more spaces at one pass in a gear.

Gear-Roughing or Stocking Cutter.—Formed cutter for roughing out gears. Frequently the teeth are irregularly nicked to break up the chip. May be single or multiple type or may be used in combination with a finishing cutter.

Sprocket Cutter.—Formed cutter for milling one space at a time in sprockets.

Multiple Sprocket-Cutters.—A single-unit formed-cutter or two or more formed-cutters made to mill two or more spaces at one pass in a sprocket.

Straddle Sprocket-Cutter.—A formed cutter for finishing one tooth at a time on roller chain sprockets.

Convex Cutter.—Formed cutter to mill a concave surface of circular contour equal to a half circle or less. Size is designated by specifying *diameter* of circular form.

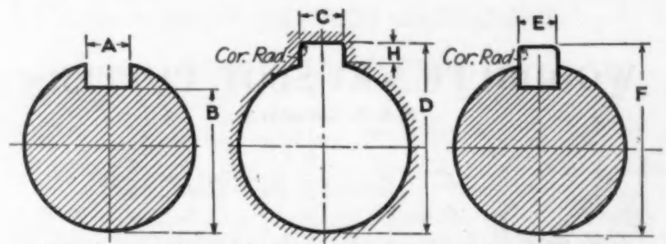
Concave Cutter.—Formed cutter shaped to mill a convex surface of circular contour equal to a half circle or less. Size is designated by specifying *diameter* of circular form.

Corner-Rounding Cutter.—Formed cutter for milling a circular corner on work up to one-quarter of a circle. May be made single or double. If single, when viewed from the small face, hand of rotation is determined as defined under that heading. Size is designated by specifying *radius* of circular form.

Spline Cutter.—These cutters may be of the single or the duplex type. The single type is a formed cutter for milling a single flute at a time in spline shafts. The duplex type are formed cutters in pairs for milling two flutes at one pass in spline shafts.

Thread-Milling Cutter.—A single cutter used for milling one thread at a time, generally worm or Acme thread type. They are customarily made 29 deg. included angle and may be either profile or formed type. In the profile type, there are two common styles, the first of which has every tooth full and complete; the second, known as the interrupted type, as every other tooth is cut away on alternate sides to afford chip clearance with the exception of one tooth which is left full and complete for gaging.

Multiple Thread-Milling Cutter.—Generally called threading hob, although *having no lead.* A shank or hole-type formed-cutter for milling threads. The length of cutting face is at least one pitch longer than the length of thread



ARBOR & KEYSEAT CUTTER BORE & KEYWAY ARBOR & KEY

Table No. 1. Keyseat, Keyway and Key Dimensions.

Diam. Arbor	Nom. Size Key (Square)	Arbor and Keyseat				Bore and Keyway[1]						Arbor and Key[1]			
		A		B		C		D	H	Cor. Rad.		E		F	
		Max.	Min.	Max.	Min.	Max.	Min.	Min.	Nom.			Max.	Min.	Max.	Min.
1/2	3/32	.0947	.0937	.4531	.4481	.106	.099	.5578	3/64	.020		.0932	.0927	.5468	.5408
5/8	1/8	.126	.125	.5625	.5575	.137	.130	.6985	1/16	1/32		.1245	.1240	.6875	.6815
3/4	1/8	.126	.125	.6875	.6825	.137	.130	.8225	1/16	1/32		.1245	.1240	.8125	.8065
7/8	1/8	.126	.125	.8125	.8075	.137	.130	.9475	1/16	1/32		.1245	.1240	.9375	.9315
1	1/4	.251	.250	.8438	.8388	.262	.255	1.104	3/32	3/64		.2495	.2490	1.094	1.088
1 1/4	5/16	.3135	.3125	1 063	1.058	.325	.318	1.385	1/8	1/16		.3120	.3115	1.375	1.369
1 1/2	3/8	.376	.375	1.281	1.276	.410	.385	1.666	5/32	1/16		.3745	.3740	1.656	1.650
1 3/4	7/16	.4385	.4375	1.500	1.495	473	.448	1.948	3/16	1/16		.4370	.4365	1.938	1.932
2	1/2	.501	.500	1.687	1.682	.535	.510	2.198	3/16	1/16		.4995	.4990	2.188	2.182
2 1/2	5/8	.626	.625	2.094	2.089	.660	.635	2.733	7/32	1/16		.6245	.6240	2.718	2.712
3	3/4	.751	.750	2.500	2.495	.785	.760	3.265	1/4	3/32		.7495	7490	3.250	3.244
3 1/2	7/8	876	.875	3.000	2.995	.910	885	3.890	3/8	3/32		.8745	.8740	3.875	3.869
4	1	1.001	1.000	3.375	3.370	1.035	1 010	4.390	3/8	3/32		9995	.9990	4.375	4.369
4 1/2	1 1/8	1.126	1.125	3.813	3.808	1.160	1.135	4.953	7/16	1/8		1 1245	1.1240	4.938	4.932
5	1 1/4	1.251	1.250	4.250	4.245	2.285	2.260	5.515	1/2	1/8		1 2495	1.2490	5.500	5.494

All dimensions in inches.

[1]Note: A difference between the overall Cutter Bore and Keyway "D" and the Arbor and Key dimension "F", of .010 inch for Arbor diameters up to 2 inches, and .015 inch on Arbor diameters larger than 2 inches, is allowed.

to be milled. Both internal and external threads may be milled, also parallel or tapered work.

Hob.—Formed milling-cutter, the teeth of which lie in a helical path about the circumferential surface of the cutter. Generally used for spur and spiral gears, wormwheels, sprocket teeth, ratchets, spline shafts, square drive shafts and similar parts.

Inserted-Tooth Cutter.—Cutter in which teeth are inserted and secured by various methods in a body of less expensive material, the object being economy in first cost and also in maintenance because of opportunity of tooth replacement. These cutters may be made with any style of relief of teeth or any method of mounting.

Inserted-Tooth Facing-Cutter.—A cutter adapted to be attached directly to spindle end or stub arbor and having inserted teeth cutting on circumferential surface and one end, similar to side mill.

Helical Mill.—Helical mills are of the profile type. They may be either hole or shank style. While most slab and shank end-mills have their peripheral teeth at a slight helix angle, the name helical mill is used to designate a high, 45 deg. or greater, helix angle of tooth. Used for slab milling or for profiling, such as cam milling and for elongating slots. Shank type with pilot end is used for elongating slots.

Intermittent-Tooth Cutter.—Formed cutter having a tooth contour of fine points such as a thread-milling cutter or hacksaw milling-cutter in which succeeding lands around the cutter carry alternately only half the necessary cutting points so staggered as to complete the full required pitch on the finished work. These cutters may be of the shank or arbor type.

Woodruff Key-Slot Cutters and Gages

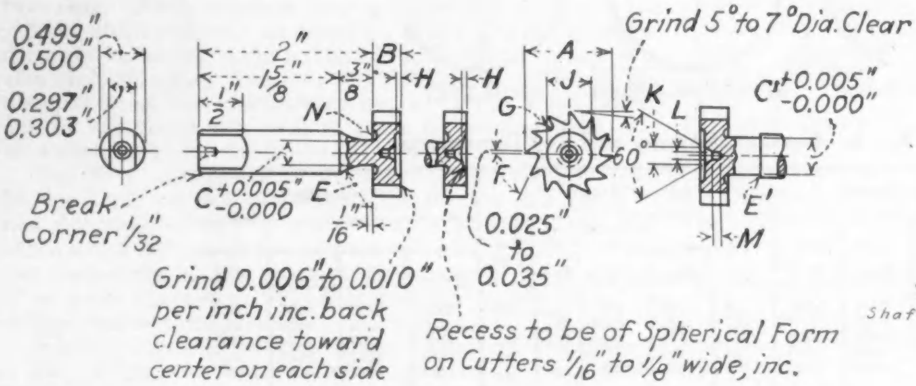
(Proposed Revision and Extension of S.A.E. Standard)

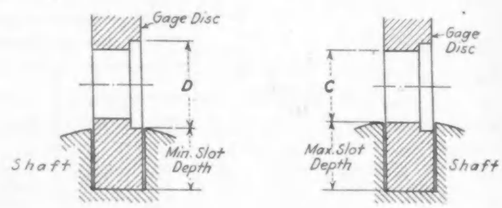
Since adoption by the Society in July, 1929, of the S.A.E. Standard for Woodruff Key-Slot Cutters, improvements in the cutter drivers make it possible to mill the key-slot more accurately in width and consequently to specify the cutter width more nearly to the slot widths. It was therefore recommended at the meeting on Dec. 19 of members of the Production Division that the present S.A.E. Standard for Woodruff Key-Slot Cutters be revised by changing the maximum cutter width B to minimum, and to make B maximum 0.0005 in. larger than the new minimum throughout the series; and to increase the back clearance from 0.004 to 0.008 in. to read 0.006 to 0.010 in. to give greater chip-clearance. It was also recommended to decrease the limit dimensions for the distance between flats on the cutter shank by 0.002 in. to fit the cutter driver. In connection with the 1/4 x 1 3/8-in. cutter, it was recommended that the neck diameter C be changed to diameter C^1 equal to 0.401; that radius E be changed to E^1 equal to 3/64 in. and that radius N be changed from 1/64 to 1/32 in. It was further recommended to include an allowance of plus 0.0005 in. on the cutter width and the minimum key-slot width in footnotes A and B following the table, which is printed on p. 42, and to change the concentricity of the tang in the footnote from 0.012 to read 0.007 in.

Following the adoption by the Society of the complete specifications for Woodruff Keys and Key-Slots and Key-Slot Cutters, the accompanying specification for Woodruff Key-Slot Gages was submitted to the Society by the General Motors Corp. Standards Department for consideration toward having a complete specification for Woodruff Keys and Key-Slots and the tools for the same. The specification was referred to the Production Division and at the Dec. 19 meeting of the Division members, it was recommended that the specification for these gages be added to the present S.A.E. Standard for Key-Slot Cutters and to supplement it with the sketches showing the application of the slot depth limits C and D as shown in the drawing at the right above the table on p. 42. It was felt that adoption of this gage specification will be very helpful to gage users and manufacturers by providing for standard gages that can be purchased from stock and assure uniformity in the practice. One point brought out was that the life of the gaging discs can be made threefold by rotating them through 120 deg. as

WOODRUFF KEY-SLOT CUTTERS

S.A.E. Standard





Method of Gaging Keyslot Depth Limits

Cutter Size	Fine Teeth		Coarse Teeth		Cutter Diameter			Width		Neck Dia.		Radii				Recess		Centerdrill	
	No. of Tth.	Ang.	No. of Tth.	Ang.	A			B								Dpth.	Dia.	Dia.	Drill
		F		F	Basic	Min.	Max.	Max.	Min.	C	C1	E	E1	G	N	H	J	K	L,M
1/16x 1/2	10	70°	8	80°	0.500	0.510	0.515	0.0625	0.0620	0.130	......	23/64	.....	1/64	1/64	1/64	1/4	3/32	1/16
3/32x 1/2	10	70°	8	80°	0.500	0.510	0.515	0.0938	0.0933	0.160	......	3/8	.....	1/64	1/64	1/64	1/4	1/8	1/16
1/8x 1/2	10	70°	8	80°	0.500	0.510	0.515	0.1250	0.1245	0.191	......	13/32	.....	1/32	1/64	1/64	1/4	1/8	1/16
3/32x 5/8	10	70°	8	80°	0.625	0.635	0.640	0.0938	0.0933	0.191	......	13/32	.....	1/32	1/64	1/64	5/16	1/8	1/16
1/8x 5/8	10	70°	8	80°	0.625	0.635	0.640	0.1250	0.1245	0.223	......	7/16	.....	1/32	1/64	1/64	5/16	1/8	1/16
5/32x 5/8	10	70°	8	80°	0.625	0.635	0.640	0.1563	0.1558	0.253	......	15/32	.....	1/32	1/64	1/64	5/16	1/8	1/16
1/8x 3/4	10	70°	8	80°	0.750	0.760	0.765	0.1250	0.1245	0.217	......	13/32	.....	1/32	1/64	1/64	3/8	1/8	1/16
5/32x 3/4	10	70°	8	80°	0.750	0.760	0.765	0.1563	0.1558	0.247	......	7/16	.....	1/32	1/64	1/64	3/8	1/8	1/16
3/16x 3/4	10	70°	8	80°	0.750	0.760	0.765	0.1873	0.1868	0.279	......	1/2	.....	1/32	1/64	1/64	3/8	1/8	1/16
5/32x 7/8	12	70°	9	80°	0.875	0.887	0.892	0.1563	0.1558	0.247	......	7/16	.....	1/32	1/64	1/32	7/16	1/8	1/16
3/16x 7/8	12	70°	9	80°	0.875	0.887	0.892	0.1873	0.1868	0.279	......	1/2	.....	1/32	1/64	1/32	7/16	1/8	1/16
1/4x 7/8	12	70°	9	80°	0.875	0.887	0.892	0.2497	0.2492	0.342	......	21/32	.....	1/32	1/64	1/32	7/16	1/8	1/16
3/16x1	12	70°	10	80°	1.000	1.012	1.017	0.1873	0.1868	0.279	......	1/2	.....	1/32	1/64	1/32	1/2	3/16	3/32
1/4x1	12	70°	10	80°	1.000	1.012	1.017	0.2497	0.2492	0.342	......	21/32	.....	1/32	1/64	1/32	1/2	3/16	3/32
5/16x1	12	70°	10	80°	1.000	1.012	1.017	0.3121	0.3116	......	0.401	.....	3/64	1/32	1/32	1/32	1/2	3/16	3/32
3/16x1 1/8	12	70°	10	80°	1.125	1.137	1.142	0.1873	0.1868	0.312	......	9/16	.....	1/32	1/64	1/32	9/16	3/16	3/32
1/4x1 1/8	12	70°	10	80°	1.125	1.137	1.142	0.2497	0.2492	0.373	......	3/4	.....	1/32	1/64	1/32	9/16	3/16	3/32
5/16x1 1/8	12	70°	10	80°	1.125	1.137	1.142	0.3121	0.3116	......	0.435	.....	1/32	1/32	1/32	1/32	9/16	3/16	3/32
1/4x1 1/4	14	70°	10	80°	1.250	1.265	1.270	0.2497	0.2492	0.373	......	3/4	.....	1/32	1/64	1/32	5/8	3/16	3/32
5/16x1 1/4	14	70°	10	80°	1.250	1.265	1.270	0.3121	0.3116	......	0.435	.....	1/32	1/32	1/32	1/32	5/8	3/16	3/32
3/8x1 1/4	14	70°	10	80°	1.250	1.265	1.270	0.3745	0.3740	......	0.467	.....	1/64	1/32	1/32	1/32	5/8	3/16	3/32
1/4x1 3/8	14	70°	10	80°	1.375	1.390	1.395	0.2497	0.2492	......	0.401	.....	3/64	1/32	1/32	1/32	11/16	3/16	3/32
5/16x1 3/8	14	70°	10	80°	1.375	1.390	1.395	0.3121	0.3116	......	0.467	.....	1/64	1/32	1/32	1/32	11/16	3/16	3/32
3/8x1 3/8	14	70°	10	80°	1.375	1.390	1.395	0.3745	0.3740	......	0.467	.....	1/64	1/32	1/32	1/32	11/16	3/16	3/32
1/4x1 1/2	16	70°	12	80°	1.500	1.515	1.520	0.2497	0.2492	......	0.435	.....	1/32	1/32	1/32	1/32	3/4	3/16	3/32
5/16x1 1/2	16	70°	12	80°	1.500	1.515	1.520	0.3121	0.3116	......	0.467	.....	1/64	1/32	1/32	1/32	3/4	3/16	3/32
3/8x1 1/2	16	70°	12	80°	1.500	1.515	1.520	0.3745	0.3740	......	0.467	.....	1/64	1/32	1/32	1/32	3/4	3/16	3/32

Cutter Diameter—A
Diameters are set with the maximum that will allow the cutter to be reground once or more, depending on the width +0.0005 in.

Cutter Width—B
Dimensions are set with minimum cutter-width at minimum keyslot-width +0.0005 in.
By adding to the maximum cutter-width the amount it will cut oversize, a slot will be produced in the majority of cases just under the maximum keyslot-dimensions.

Concentricity
0.002 in. by indicator reading.
Concentricity of tang 0.007 in. by indicator reading.

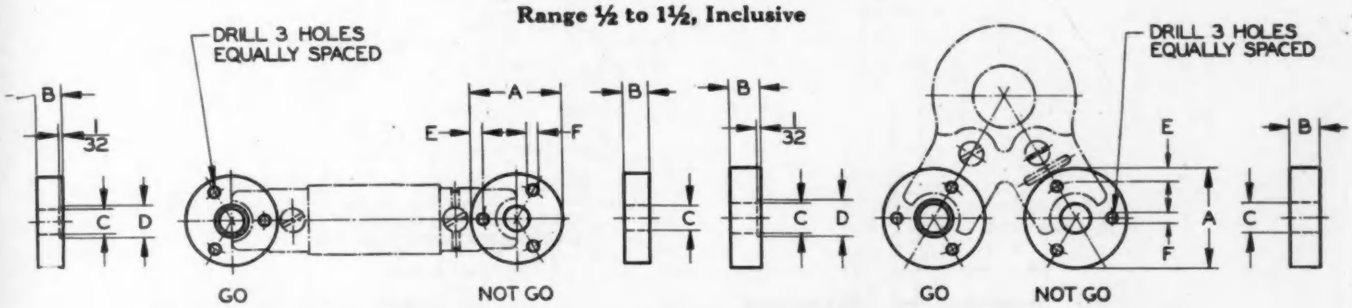
Parallelism
Sides to run true to within 0.0005 in.

Marking and Manufacture
Size and manufacturer's name or trademark to be marked on the shank. All burrs must be removed.

Material
High-speed steel or equivalent, hardened.

WOODRUFF KEYSLOT GAGE DISCS

Range ½ to 1½, Inclusive



Woodruff Key Size	General Dimensions												
	Go Disc								Not Go Disc				
	Depth		A	B	C	D	E	F	A	B	C	E	F
	Min.	Max.											
1/16 x 1/2	.1668	.1718	.5000	.0615	.1564	.1664	.0781	1/16	.5000	.0630	3/16	.0781	1/16
3/32 x 1/2	.1511	.1561	.5000	.0928	.1878	.1978	.0781	1/16	.5000	.0943	3/16	.0781	1/16
1/8 x 1/2	.1355	.1405	.5000	.1240	.2190	.2290	.0781	1/16	.5000	.1255	3/16	.0781	1/16
3/32 x 5/8	.1981	.2031	.6250	.0928	.2188	.2288	.0938	3/32	.6250	.0943	7/32	.0938	3/32
1/8 x 5/8	.1825	.1875	.6250	.1240	.2500	.2600	.0938	3/32	.6250	.1255	7/32	.0938	3/32
5/32 x 5/8	.1669	.1719	.6250	.1553	.2812	.2912	.0938	3/32	.6250	.1568	7/32	.0938	3/32
1/8 x 3/4	.2455	.2505	.7500	.1240	.2490	.2590	.0938	3/32	.7500	.1255	1/4	.0938	3/32
5/32 x 3/4	.2299	.2349	.7500	.1553	.2802	.2902	.0938	3/32	.7500	.1568	1/4	.0938	3/32
3/16 x 3/4	.2143	.2193	.7500	.1863	.3114	.3214	.0938	3/32	.7500	.1880	1/4	.0938	3/32
5/32 x 7/8	.2919	.2969	.8750	.1553	.2812	.2912	.1094	1/8	.8750	.1568	9/32	.1094	1/8
3/16 x 7/8	.2763	.2813	.8750	.1863	.3124	.3224	.1094	1/8	.8750	.1880	9/32	.1094	1/8
1/4 x 7/8	.2450	.2500	.8750	.2487	.3750	.3850	.1094	1/8	.8750	.2505	9/32	.1094	1/8
3/16 x 1	.3393	.3443	1.0000	.1863	.3114	.3214	.1250	1/8	1.0000	.1880	5/16	.1250	1/4
1/4 x 1	.3080	.3130	1.0000	.2487	.3740	.3840	.1250	1/8	1.0000	.2505	5/16	.1250	1/8
5/16 x 1	.2768	.2818	1.0000	.3111	.4364	.4464	.1250	1/8	1.0000	.3130	5/16	.1250	1/8
3/16 x 1 1/8	.3853	.3903	1.1250	.1863	.3444	.3544	.1562	1/8	1.1250	.1880	11/32	.1562	1/8
1/4 x 1 1/8	.3540	.3590	1.1250	.2487	.4070	.4170	.1562	1/8	1.1250	.2505	11/32	.1562	1/8
5/16 x 1 1/8	.3228	.3278	1.1250	.3111	.4694	.4794	.1562	1/8	1.1250	.3130	11/32	.1562	1/8
1/4 x 1 1/4	.4170	.4220	1.2500	.2487	.4060	.4160	.1719	1/8	1.2500	.2505	3/8	.1719	1/8
5/16 x 1 1/4	.3858	.3908	1.2500	.3111	.4684	.4784	.1719	1/8	1.2500	.3130	3/8	.1719	1/8
3/8 x 1 1/4	.3545	.3595	1.2500	.3735	.5310	.5410	.1719	1/8	1.2500	.3755	3/8	.1719	1/8
1/4 x 1 3/8	.4640	.4690	1.3750	.2487	.4370	.4470	.1875	1/8	1.3750	.2505	7/16	.1875	1/8
5/16 x 1 3/8	.4328	.4378	1.3750	.3111	.4994	.5094	.1875	1/8	1.3750	.3130	7/16	.1875	1/8
3/8 x 1 3/8	.4015	.4065	1.3750	.3735	.5620	.5720	.1875	1/8	1.3750	.3755	7/16	.1875	1/8
1/4 x 1 1/2	.5110	.5160	1.5000	.2487	.4680	.4780	.2187	1/8	1.5000	.2505	1/2	.2187	1/8
5/16 x 1 1/2	.4798	.4848	1.5000	.3111	.5304	.5404	.2187	1/8	1.5000	.3130	1/2	.2187	1/8
3/8 x 1 1/2	.4485	.4535	1.5000	.3735	.5930	.6030	.2187	1/8	1.5000	.3755	1/2	.2187	1/8

GENERAL SPECIFICATIONS

1. Outside Diameter—A.
Dimensions are basic key diameters.
All sizes—Go: Basic plus .0002, minus .000
Not go: Basic plus or minus .001

2. Width—B.
Dimensions are set for minimum keyslot width for "Go" disc and maximum keyslot width for "Not Go" disc.
All Sizes—Go: Size plus .0002, minus .000
Not Go: Size plus .0002, minus .000

3. Diameter—C.
Dimensions are set for maximum keyslot depth for "Go" disc. Drilled holes are specified for "Not Go" disc.
All sizes—Go: Size plus .0004, minus .000
Not Go: Size plus or minus .005

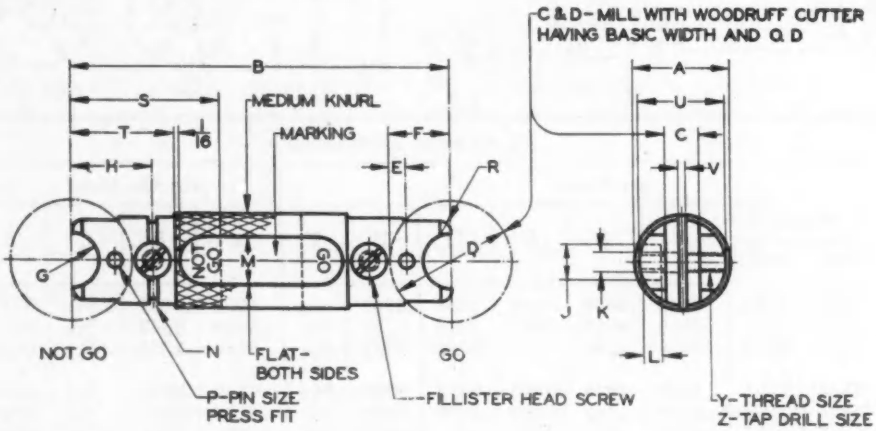
4. Diameter—D.
Dimensions are set for minimum keyslot depth.
All sizes—Size plus .0004, minus .000

5. Material
Oil hardening steel, or equivalent, hardened and ground as specified.

6. Rockwell Hardness

7. Marking and Manufacture
Go: Minimum keyslot width to be engraved or acid etched on side of disc. All burrs must be removed.
Not Go: Maximum keyslot width to be engraved or acid etched on side of disc. All burrs must be removed.

8. Finish
Grind all surfaces on "Go" disc. Grind flat surfaces only on "Not Go" disc. Break corners on all discs 1/64 to 1/32 × 45° according to width.
Ground surfaces may be lapped if this is according to manufacturer's practice, it being understood that if grinding alone is done, the finished surface must be acceptable to user.



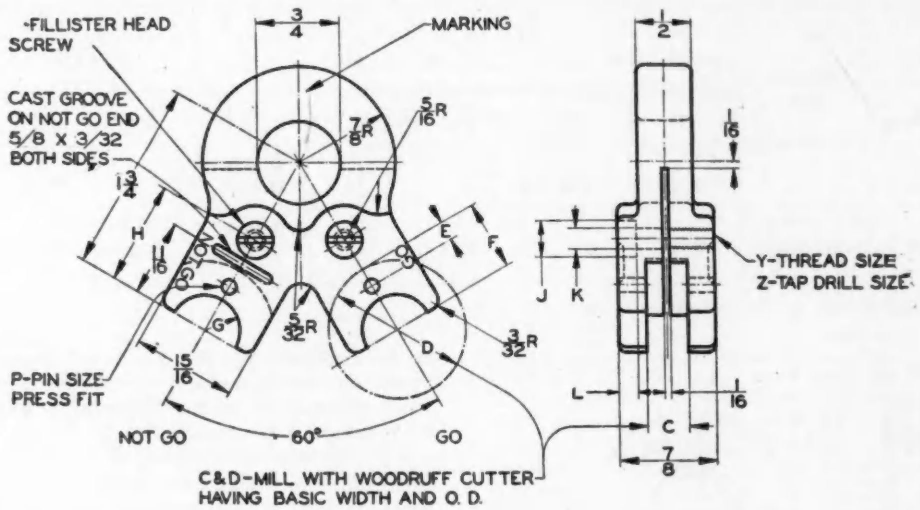
Key Size	General Dimensions																				
	A	B	C	D	E	F	G	H	J	K	L	M	N	P	R	S	T	U	V	Y Size	Y Pitch Dia.	Z
1/16 x 1/2	1/2	2 3/4	1/16	1/2	.0781	1/4	7/64	3/8	2420	# 27(.1440)	1/8	1/4	1/16 x 1/32	1/16	5/64	15/16	9/16	7/16	1/32	# 6-32	.1177-.1196	# 37 (.1040)
3/32 x 1/2	1/2	2 3/4	3/32	1/2	.0781	1/4	7/64	3/8	.2420	# 27(.1440)	1/8	1/4	1/16 x 1/32	1/16	5/64	15/16	9/16	7/16	1/32	# 6-32	.1177-.1196	# 37 (.1040)
1/8 x 1/2	1/2	2 3/4	1/8	1/2	.0781	1/4	7/64	3/8	.2420	# 27(.1440)	1/8	1/4	1/16 x 1/32	1/16	5/64	15/16	9/16	7/16	1/32	# 6-32	.1177-.1196	# 37 (.1040)
3/32 x 5/8	5/8	2 7/8	3/32	5/8	.0938	5/16	9/64	15/32	21/64	# 7 (.2010)	11/64	5/16	1/16 x 1/32	3/32	5/64	1	5/8	9/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
1/8 x 5/8	5/8	2 7/8	1/8	5/8	.0938	5/16	9/64	15/32	21/64	# 7 (.2010)	11/64	5/16	1/16 x 1/32	3/32	5/64	1	5/8	9/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
5/32 x 5/8	5/8	2 7/8	5/32	5/8	.0938	5/16	9/64	15/32	21/64	# 7 (.2010)	11/64	5/16	1/16 x 1/32	3/32	5/64	1	5/8	9/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
1/8 x 3/4	3/4	3	1/8	3/4	.0938	3/8	3/16	17/32	21/64	# 7 (.2010)	11/64	3/8	3/32 x 3/64	3/32	3/32	1 1/16	11/16	11/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
5/32 x 3/4	3/4	3	5/32	3/4	.0938	3/8	3/16	17/32	21/64	# 7 (.2010)	11/64	3/8	3/32 x 3/64	3/32	3/32	1 1/16	11/16	11/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
3/16 x 3/4	3/4	3	3/16	3/4	.0938	3/8	3/16	17/32	21/64	# 7 (.2010)	11/64	3/8	3/32 x 3/64	3/32	3/32	1 1/16	11/16	11/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
5/32 x 7/8	3/4	3 1/8	5/32	7/8	.1094	7/16	7/32	19/32	21/64	# 7 (.2010)	11/64	3/8	3/32 x 3/64	1/8	3/32	1 3/16	3/4	11/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
3/16 x 7/8	3/4	3 1/8	3/16	7/8	.1094	7/16	7/32	19/32	21/64	# 7 (.2010)	11/64	3/8	3/32 x 3/64	1/8	3/32	1 3/16	3/4	11/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
1/4 x 7/8	3/4	3 1/8	1/4	7/8	.1094	7/16	7/32	19/32	21/64	# 7 (.2010)	11/64	3/8	3/32 x 3/64	1/8	3/32	1 3/16	3/4	11/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
3/16 x 1	7/8	3 3/8	3/16	1	.1250	1/2	1/4	11/16	21/64	# 7 (.2010)	11/64	7/16	1/8 x 1/16	1/8	3/32	1 1/4	7/8	13/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
1/4 x 1	7/8	3 3/8	1/4	1	.1250	1/2	1/4	11/16	21/64	# 7 (.2010)	11/64	7/16	1/8 x 1/16	1/8	3/32	1 1/4	7/8	13/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
5/16 x 1	7/8	3 3/8	5/16	1	.1250	1/2	1/4	11/16	21/64	# 7 (.2010)	11/64	7/16	1/8 x 1/16	1/8	3/32	1 1/4	7/8	13/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
3/16 x 1 1/8	7/8	3 1/2	3/16	1 1/8	.1562	9/16	1/4	3/4	21/64	# 7 (.2010)	11/64	7/16	1/8 x 1/16	1/8	3/32	1 3/8	15/16	13/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
1/4 x 1 1/8	7/8	3 1/2	1/4	1 1/8	.1562	9/16	1/4	3/4	21/64	# 7 (.2010)	11/64	7/16	1/8 x 1/16	1/8	3/32	1 3/8	15/16	13/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)
5/16 x 1 1/8	7/8	3 1/2	5/16	1 1/8	.1562	9/16	1/4	3/4	21/64	# 7 (.2010)	11/64	7/16	1/8 x 1/16	1/8	3/32	1 3/8	15/16	13/16	1/16	# 10-32	.1697-.1716	# 23 (.1540)

For key sizes 1/16 x 1/2 to and including 1/8 x 1/2 use 6-32, 5/16-in. long fillister-head screw.

For key sizes 3/32 x 5/8 to and including 5/32 x 5/8 use 10-32, 3/8-in. long fillister-head screw.

For key sizes 1/8 x 3/4 to and including 1/4 x 7/8 use 10-32, 1/2-in. long fillister-head screw.

For key sizes 3/16 x 1 to and including 5/16 x 1 1/8 use 10-32, 5/8-in. long fillister-head screw.



Key Size	General Dimensions																				
	A	B	C	D	E	F	G	H	J	K	L	M	N	P	R	S	T	U	V	Y Size	Y Pitch Dia.	Z
1/4 x 1 1/4	—	—	1/4	1 1/4	.1719	5/8	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
5/16 x 1 1/4	—	—	5/16	1 1/4	.1719	5/8	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
3/8 x 1 1/4	—	—	3/8	1 1/4	.1719	5/8	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
1/4 x 1 3/8	—	—	1/4	1 3/8	.1875	11/16	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
5/16 x 1 3/8	—	—	5/16	1 3/8	.1875	11/16	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
3/8 x 1 3/8	—	—	3/8	1 3/8	.1875	11/16	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
1/4 x 1 1/2	—	—	1/4	1 1/2	.2187	3/4	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
5/16 x 1 1/2	—	—	5/16	1 1/2	.2187	3/4	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)
3/8 x 1 1/2	—	—	3/8	1 1/2	.2187	3/4	5/16	15/16	21/64	#7 (.2010)	11/64	—	—	1/8	—	—	—	—	—	#10-32	.1697-.1716	#23 (.1540)

GENERAL SPECIFICATIONS

1. **Material**
   Range 1/2 to 1 1/8 Incl., Aluminum or duralumin rod.
   Range 1 1/4 to 1 1/2 Incl., Aluminum casting.
   Pin "P," Drill rod.
2. **Marking and Manufacture**
   Basic key size and manufacturer's name or trade mark to be stamped as indicated. All burrs must be removed.

For key sizes 1/4 x 1 1/4 to and including 3/8 x 1 1/2 use 10-32, 3/4-in. long fillister-head screw.

provided for by the three holding-pin holes shown in the discs.

The Production Division therefore submits this report with the recommendation that it be approved and adopted by the Society as S.A.E. Standard for Woodruff Key-Slot Cutters and Gages. At the time of printing the report, it was subject to letter-ballot confirmation by the Production Division under the Standards Regulations of the Society.

Plug and Ring Gages

(Proposed Revision of S.A.E. Recommended Practice)

At a meeting in New York City in November, 1926, the Gage Section of the Ordnance Department and the U. S. Army decided on the standardization for cylindrical and thread plug gages and ring gages of the sizes most commonly used in interchangeable manufacture. The first report of a committee that had been organized for cylindrical plug and thread gages was submitted to the Society and adopted by it in August, 1928, as S.A.E. Production Recommended Practice. Revisions in the report have since been made and submitted to the Production Division of the Society's Standards Committee and approved for submission to the Society for adoption. The original recommendation and subsequent revisions have been approved by the gage manufacturers and many users and it is felt that the refinements in and extensions to the report should be adopted by the Society to keep the S.A.E. Recommended Practice uptodate. Most of the recommendations submitted herewith are minor corrections that have been made by the originating technical sub-committee of the American Gage Design Committee, the principal change being to extend the optional range given in the table on p. 3 of the S.A.E. Production Recommended Practice from above 2.010 to and including 2.510 for the cylindrical plugs. The Production Division therefore recommends that the accompanying revisions be approved and adopted by the Society. When printed this report was subject to a confirming letter-ballot of the Production Division members under the Society's Standards Regulations.

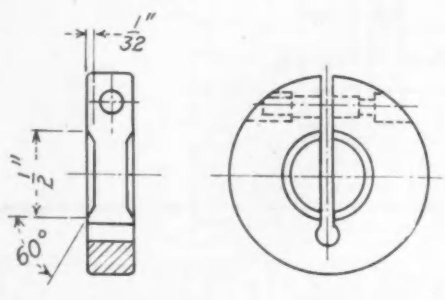
Referring to the S.A.E. Production Recommended Practice for Plug and Ring Gages on p. 2, the depth of groove in the handles for the Nos. 000, 00 and 0 handle sizes are changed to 0.172-0.177; 0.235-0.240, and 0.279-0.302 respectively.

Extend Table 2 on p. 3 for Cylindrical Plugs by adding the dimensions to make table read as printed on p. 46.

Revise the illustrations on p. 6 by including a drawing for the No. 0 to No. 6 blanks only, showing a 1/2-in. counterbore 1/32 in. deep on each side, specifying chamfer on all blanks 1/2 to 1 thread deep and omitting the groove from the thick blanks. In Table 5 following these illustrations, change the thickness of the No. 0 to No. 6 blank from

Range in Size		Handle Size	Go		All			Not Go		Progressive	
Above	Including		*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*
1.510[a]	2.010	5	3¼	1⅞	1	⅜	0.808 0.810	2¼	⅞	4¼	2⅞
2.010[a]	2.510	5	3⅜	2	1	⅜	0.808 0.810	2¼	⅞	4⅜	3

[a] Optional range applies only to the cylindrical plug-gages.



Bore on No. 0 to No. 6 Blank Only

13/64 to ¼ in. and the range of the largest size blanks from 4⅛ to 4¾ to read 4⅛ to 4½.

It is also recommended that the optional handles of the round knurled type shown on p. 2 of the S.A.E. Recommended Practice be discontinued as the standard hexagon type is preferred and is the type carried in stock by gage manufacturers for the great majority of purchasers.

Taps—Cut and Ground Threads

(Proposed American Standard and Revision of S.A.E. Standard)

The S.A.E. Standard for Taps with Cut and Ground Threads that was originally adopted by the Society in February, 1927, is substantially the same as the report of the Sectional Committee on Small Tools and Machine-Tool Elements that has been in progress for about 2 years with the exception that in the Sectional Committee's report the tables for tapper taps with cut and ground threads include 1¾ in.-5 pitch and 2 in.-4½ pitch taps and also tables for nut taps—cut and ground threads and pulley taps—cut threads which are not included in the S.A.E. Standard. The Sectional Committee has submitted its report on Taps dated November, 1929, to the Society as one of the sponsors for the Sectional Committee for its approval and adoption as American Standard. The report was referred to the Production Division of the Society's Standards Committee and was discussed at some length at the meeting of the members of the Division held on Dec. 19 last. A number of minor differences were noted in the notes following the tables of tap dimensions, relating to the range of tap sizes to which tolerances apply, but the Division members felt that the footnotes in the Sectional Committee's report should be approved as revisions in the corresponding S.A.E. notes. It was noted that the pilot diameter for the tapper taps were not included in the Sectional Committee's report and that the land width on the flutes is not given although the latter was not considered so important for inclusion in a general standard.

The Production Division therefore submits the report of the Sectional Committee for approval and adoption as American Standard and also in revision of the present S.A.E. Standard with pilot diameters for the tapper taps to be included by the Sectional Committee in its report, based on the formula

Pilot diameter = Minor diameter + 0.0000 in., — 0.0050 in. as given in the present S.A.E. Standard.

The Division also recommends that when revision of its report is next under consideration by the Sectional Committee, tables for straight pipe-thread taps and bent-shank tapper-taps similar to the standards adopted by the General Motors Corp., be included in the report. The Division further recommends that to avoid duplicate publication of the standard, the Society continue to publish only its tap standards for straight pipe-thread taps and spark-plug taps, as printed on p. 6 of the S.A.E. Production Standard, corrected and reprinted as of August, 1929, with the understanding that copies of the Sectional Committee's report be furnished to all members of the Society who receive the S.A.E. Production Standards. These recommendations of the Division at the time of printing them were subject to confirming letter-ballot by the Production Division as a quorum was not present at the Division meeting.

Cut and Ground Taps

General.—In developing this standard for Cut and Ground Thread Taps it is anticipated that additional styles will be added from time to time. The present standard covers the thread and general dimensions together with working tolerances and basic formula for the following types of taps: Machine Screw Taps, Hand Taps, Tapper Taps, Nut Taps and Pulley Taps.

Size of Tapped Hole.—On account of the wide variation in manufacturing methods and shop conditions, the size of the tapped hole cannot be guaranteed.

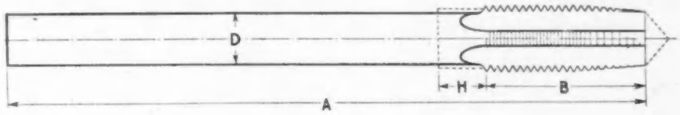
Cut-thread taps made to the specifications when used under normal conditions, should, in the majority of cases, produce holes within the American Standard Class 2 tolerance.

Ground-thread taps made to the specifications when used under normal conditions, should, in the majority of cases, produce holes within the American Standard Class 3 tolerance.

Marking.—All taps shall be clearly marked with the size of tap and the number of threads per inch together with the letters *NC*, indicating the American National Coarse Thread Series; *NF*, indicating the American National Fine Thread Series; or *NS*, indicating the American National form of thread with a special number of threads per inch. In addition to the above, all ground thread taps shall be marked with the letter *G*. These markings supercede the old thread series designations: USS, USF, SAE and ASME.

Form of Thread.—The form of thread profile as specified for the taps shown in this standard is known as the American National Standard Form. The basic angle of thread between the sides of thread measured in an axial plane is 60 deg. The line bisecting this 60-deg. angle is perpendicular to the axis of the screw thread. The basic depth of the thread form h is found as follows:

$$h = 0.649519 \times p = 0.649519/n$$



Tapper Taps
Cut Thread

Size	Threads per Inch		Pitch Diameter			Major Diameter			General Dimensions			
	NC	NF	Basic	Min.	Max.	Basic	Min.	Max.	A	B	D	H[1]
1¾	5		1.6201	1.6216	1.6256	1.7500	1.7575	1.7630	15	4½	1.484	
2	4½		1.8557	1.8572	1.8612	2.0000	2.0085	2.0140	15	4½	1.705	

All dimensions are in inches.
Taps ⅜ inch and smaller have external center on thread end.
Taps 7/16 inch and larger have internal center on thread end.
[1]NOTE: A nut guide of length H approximately equal in diameter to the basic root diameter may be furnished on taps having threads NF Standard or finer.
Tapper taps are furnished with plain round shank unless otherwise ordered. For other styles of shanks furnished, consult latest trade catalogs.

Tolerances

LEAD	plus or minus .003 in. per inch of thread.
LENGTH OVERALL	¼-1 in. incl., plus or minus ⅛ in. 1⅛-2 in. incl., plus or minus 3/16 in.
LENGTH OF THREAD	¼-½ in. incl., plus or minus 1/16 in. 9/16-1½ in. incl., plus or minus 3/32 in. 1¾-2 in. incl., plus or minus ⅛ in.
DIAMETER OF SHANK	¼-½ in. incl., size to size minus .005 in. 9/16-1 in. incl., size to size minus .006 in. 1⅛-2 in. incl., size to size minus .008 in.
HALF ANGLE	4½-5 threads per inch, plus or minus 35 min. 6-9 threads per inch, plus or minus 40 min. 10-28 threads per inch, plus or minus 45 min.
FULL ANGLE	4½-5 threads per inch, 53 min. 6-9 threads per inch, 60 min. 10-28 threads per inch, 68 min.
MAJOR DIAMETER	¼-2 in. incl., maximum = basic + $\sqrt{\frac{\text{basic diam.}}{100}}$ ¼-⅜ in. incl., minimum = maximum minus .0025 in. 7/16-⅝ in. incl., minimum = maximum minus .003 in. ¾-1 in. incl., minimum = maximum minus .004 in. 1⅛-1½ in. incl., minimum = maximum minus .0045 in. 1¾-2 in. incl., minimum = maximum minus .0055 in.

Tapper Taps
Ground Thread

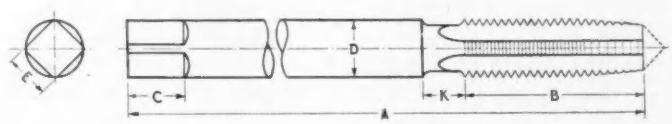
Size	Threads per Inch		Pitch Diameter			Major Diameter			General Dimensions			
	NC	NF	Basic	Min.	Max.	Basic	Min.	Max.	A	B	D	H[1]
1¾	5		1.6201	1.6216	1.6236	1.7500	1.7620	1.7650	15	4½	1.484	
2	4½		1.8557	1.8572	1.8592	2.0000	2.0130	2.0160	15	4½	1.705	

All dimensions are in inches. All taps have internal center on thread end.
[1]NOTE: A nut guide of length H, approximately equal in diameter to the basic root diameter may be furnished on taps having threads NF Standard, or finer.
Tapper taps are furnished with plain round shank unless otherwise ordered. For other styles of shanks furnished, consult latest trade catalogs.

Tolerances

LEAD	plus or minus .0005 in. per inch of thread.
LENGTH OVERALL	¼-1 in. incl., plus or minus ⅛ in. 1⅛-2 in. incl., plus or minus 3/16 in.
LENGTH OF THREAD	¼-½ in. incl., plus or minus 1/16 in. 9/16-1½ in. incl., plus or minus 3/32 in. 1¾-2 in. incl., plus or minus ⅛ in.
DIAMETER OF SHANK	¼-½ in. incl., size to size minus .005 in. 9/16-1 in. incl., size to size minus .006 in. 1⅛-2 in. incl., size to size minus .008 in.
HALF ANGLE	4½-5 threads per inch, plus or minus 20 min. 6-9 threads per inch, plus or minus 25 min. 10-28 threads per inch, plus or minus 30 min.
MAJOR DIAMETER	4½-5 threads per inch, maximum = basic plus 35 per cent truncation 6-12 threads per inch, maximum = basic plus 40 per cent truncation 13-28 threads per inch, maximum = basic plus 45 per cent truncation ¼-⅝ in. incl., minimum = maximum minus .0015 in. ¾-1 in. incl., minimum = maximum minus .002 in. 1⅛-1½ in. incl., minimum = maximum minus .0025 in. 1¾-2 in. incl., minimum = maximum minus .003 in.

Pulley Taps
Cut Thread



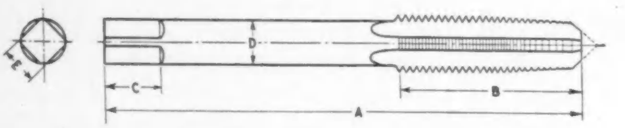
Size	Threads per Inch	Pitch Diameter			Major Diameter			General Dimensions					
	NC	Basic	Min.	Max.	Basic	Min.	Max.	A	B	C	D	E	K
¼	20	0.2175	0.2180	0.2200	0.2500	0.2525	0.2550	6, 8	1	5/16	0.255	0.191	⅜
5/16	18	2764	.2769	.2789	.3125	.3155	.3180	6, 8	1⅛	⅜	.318	.238	⅜
⅜	16	.3344	.3349	.3369	.3750	.3785	.3810	6, 8, 10, 12, 14	1¼	7/16	.381	.286	⅜
7/16	14	.3911	.3916	.3941	.4375	.4410	.4440	6, 8, 10, 12, 14	1 7/16	½	.444	.333	7/16
½	13	.4500	.4505	.4530	.5000	.5040	.5070	6, 8, 10, 12, 14	1 21/32	9/16	.507	.380	½
9/16	12	.5084	.5089	.5114	.5625	.5670	.5700	6, 8, 10, 12, 14	1 21/32	⅝	.570	.427	9/16
⅝	11	.5660	.5665	.5690	.6250	.6300	.6330	6, 8, 10, 12, 14	1 13/16	11/16	.633	.475	⅝
¾	10	.6850	.6855	.6885	.7500	.7550	.7590	10, 12, 14	2	¾	.759	.569	¾

All dimensions are in inches.
Taps 5/16 inch and smaller have external center on thread end.
Taps ⅜ inch and larger have internal center on thread end.

Tolerances

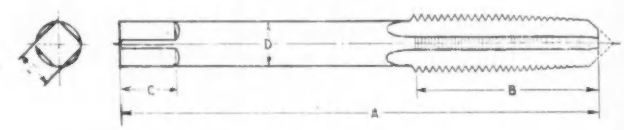
LEAD	plus or minus .003 in. per inch of thread
LENGTH OVERALL	plus or minus 1/16 in.
LENGTH OF THREAD	plus or minus 1/16 in.
LENGTH OF SQUARE	plus or minus 1/32 in.
DIAMETER OF SHANK	size to size minus .005 in.
SIZE OF SQUARE	¼-½ in. incl., size to size minus .004 in. 9/16-¾ in. incl., size to size minus .006 in.
HALF ANGLE	plus or minus 45 min.
FULL ANGLE	plus or minus 68 min.
MAJOR DIAMETER	¼-¾ in. incl., maximum = basic + $\sqrt{\frac{\text{basic diam.}}{100}}$ ¼-⅜ in. incl., minimum = maximum minus .0025 in. 7/16-⅝ in. incl., minimum = maximum minus .003 in. ¾ in. minimum = maximum minus .004 in.

Nut Taps
Cut Thread



Size	Threads per Inch		Pitch Diameter			Major Diameter			General Dimensions				
	NC	NF	Basic	Min.	Max.	Basic	Min.	Max.	A	B	C	D	E
1/4	20		0.2175	0.2180	0.2220	0.2500	0.2525	0.2550	5	1 5/8	9/16	0.185	0.139
1/4		28	.2268	.2273	.2288	.2500	.2525	.2550	5	1 1/4	9/16	.185	.139
5/16	18		.2764	.2769	.2789	.3125	.3155	.3180	5 1/2	1 13/16	5/8	.240	.180
5/16		24	.2854	.2859	.2874	.3125	.3155	.3180	5 1/2	1 3/8	5/8	.240	.180
3/8	16		.3344	.3349	.3369	.3750	.3785	.3810	6	2	11/16	.294	.220
3/8		24	.3479	.3484	.3499	.3750	.3785	.3810	6	1 1/2	11/16	.294	.220
7/16	14		.3911	.3916	.3941	.4375	.4410	.4440	6 1/2	2 3/8	3/4	.345	.259
7/16		20	.4050	.4055	.4075	.4375	.4410	.4440	6 1/2	1 3/4	3/4	.345	.259
1/2	13		.4500	.4505	.4530	.5000	.5040	.5070	7	2 1/2	7/8	.400	.300
1/2		20	.4675	.4680	.4700	.5000	.5040	.5070	7	1 7/8	7/8	.400	.300
9/16	12		.5084	.5089	.5114	.5625	.5670	.5700	7 1/2	2 3/4	7/8	.450	.337
9/16		18	.5264	.5269	.5289	.5625	.5670	.5700	7 1/2	2	7/8	.450	.337
5/8	11		.5660	.5665	.5690	.6250	.6300	.6330	8	3	15/16	.503	.377
5/8		18	.5889	.5894	.5914	.6250	.6300	.6330	8	2 1/4	15/16	.503	.377
3/4	10		.6850	.6855	.6885	.7500	.7550	.7590	9	3 1/4	1	.616	.462
3/4		16	.7094	.7099	.7124	.7500	.7550	.7590	9	2 1/2	1	.616	.462
7/8	9		.8028	.8038	.8068	.8750	.8805	.8845	10	3 5/8	1 1/16	.727	.545
7/8		14	.8286	.8296	.8321	.8750	.8805	.8845	10	2 3/4	1 1/16	.727	.545
1	8		.9188	.9198	.9228	1.0000	1.0060	1.0100	11	4	1 1/8	.834	.625
1		14	.9536	.9546	.9571	1.0000	1.0060	1.0100	11	3	1 1/8	.834	.625
1 1/8	7		1.0322	1.0332	1.0367	1.1250	1.1310	1.1355	11 1/2	4 3/4	1 1/4	.933	.700
1 1/8		12	1.0709	1.0719	1.0749	1.1250	1.1310	1.1355	11 1/2	3 1/2	1 1/4	.933	.700
1 1/4	7		1.1572	1.1582	1.1617	1.2500	1.2565	1.2610	12	4 3/4	1 5/16	1.058	.793
1 1/4		12	1.1959	1.1969	1.1999	1.2500	1.2565	1.2610	12	3 1/2	1 5/16	1.058	.793
1 1/2	6		1.3917	1.3927	1.3962	1.5000	1.5075	1.5120	13	5 3/8	1 1/2	1.278	.958
1 1/2		12	1.4459	1.4469	1.4499	1.5000	1.5075	1.5120	13	4	1 1/2	1.278	.958

Nut Taps
Ground Thread



Size	Threads per Inch		Pitch Diameter			Major Diameter			General Dimensions				
	NC	NF	Basic	Min.	Max.	Basic	Min.	Max.	A	B	C	D	E
1/4	20		0.2175	0.2180	0.2185	0.2500	0.2535	0.2550	5	1 5/8	9/16	0.185	0.139
1/4		28	.2268	.2273	.2278	.2500	.2520	.2535	5	1 1/4	9/16	.185	.139
5/16	18		.2764	.2769	.2774	.3125	.3165	.3180	5 1/2	1 13/16	5/8	.240	.180
5/16		24	.2854	.2859	.2864	.3125	.3150	.3165	5 1/2	1 3/8	5/8	.240	.180
3/8	16		.3344	.3349	.3354	.3750	.3795	.3810	6	2	11/16	.294	.220
3/8		24	.3479	.3484	.3489	.3750	.3775	.3790	6	1 1/2	11/16	.294	.220
7/16	14		.3911	.3916	.3921	.4375	.4425	.4440	6 1/2	2 3/8	3/4	.345	.259
7/16		20	.4050	.4055	.4060	.4375	.4410	.4425	6 1/2	1 3/4	3/4	.345	.259
1/2	13		.4500	.4505	.4510	.5000	.5055	.5070	7	2 1/2	7/8	.400	.300
1/2		20	.4675	.4680	.4685	.5000	.5035	.5050	7	1 7/8	7/8	.400	.300
9/16	12		.5084	.5089	.5094	.5625	.5685	.5700	7 1/2	2 3/4	7/8	.450	.337
9/16		18	.5264	.5269	.5274	.5625	.5665	.5680	7 1/2	2	7/8	.450	.337
5/8	11		.5660	.5665	.5676	.6250	.6315	.6330	8	3	15/16	.503	.377
5/8		18	.5889	.5894	.5902	.6250	.6290	.6305	8	2 1/4	15/16	.503	.377
3/4	10		.6850	.6855	.6886	.7500	.7570	.7590	9	3 1/4	1	.616	.462
3/4		16	.7094	.7099	.7107	.7500	.7540	.7560	9	2 1/2	1	.616	.462
7/8	9		.8028	.8038	.8050	.8750	.8830	.8850	10	3 5/8	1 1/16	.727	.545
7/8		14	.8286	.8296	.8305	.8750	.8795	.8815	10	2 3/4	1 1/16	.727	.545
1	8		.9188	.9198	.9212	1.0000	1.0090	1.0110	11	4	1 1/8	.834	.625
1		14	.9536	.9546	.9555	1.0000	1.0045	1.0065	11	3	1 1/8	.834	.625
1 1/8	7		1.0322	1.0332	1.0347	1.1250	1.1345	1.1370	11 1/2	4 3/4	1 1/4	.933	.700
1 1/8		12	1.0709	1.0719	1.0729	1.1250	1.1300	1.1325	11 1/2	3 1/2	1 1/4	.933	.700
1 1/4	7		1.1572	1.1582	1.1597	1.2500	1.2595	1.2620	12	4 3/4	1 5/16	1.058	.793
1 1/4		12	1.1959	1.1969	1.1979	1.2500	1.2550	1.2575	12	3 1/2	1 5/16	1.058	.793
1 1/2	6		1.3917	1.3927	1.3945	1.5000	1.5115	1.5140	13	5 3/8	1 1/2	1.278	.958
1 1/2		12	1.4459	1.4469	1.4479	1.5000	1.5050	1.5075	13	4	1 1/2	1.278	.958

Nut Taps
Cut Thread

Size	Threads per Inch	Pitch Diameter			Major Diameter			General Dimensions				
	NC	Basic	Min.	Max.	Basic	Min.	Max.	A	B	C	D	E
1 3/4	5	1.6201	1.6216	1.6256	1.7500	1.7575	1.7630	14	5 1/2	1 5/8	1.484	1.113
2	4 1/2	1.8557	1.8572	1.8612	2.0000	2.0085	2.0140	15	6 1/8	1 3/4	1.705	1.279
2 1/4	4 1/2	2.1057	2.1072	2.1117	2.2500	2.2590	2.2650	16	6 1/8	1 7/8	1.953	1.465
2 1/2	4	2.3376	2.3396	2.3441	2.5000	2.5100	2.5160	17	6 7/8	2	2.167	1.625

All dimensions are in inches.
Taps 1/2 inch and smaller have external center on thread end.
Taps 9/16 inch and larger have internal center on thread end.

Tolerances

LEAD — plus or minus .003 inch per inch of thread.

LENGTH OVERALL — 1/4-1 in. incl., plus or minus 1/16 in.
1 1/8-2 1/2 in. incl., plus or minus 3/32 in.

LENGTH OF THREAD — 1/4-1/2 in. incl., plus or minus 1/16 in.
9/16-1 1/2 in. incl., plus or minus 3/32 in.
1 3/4-2 1/2 in. incl., plus or minus 1/8 in.

LENGTH OF SQUARE — 1/4-5/8 in. incl., plus or minus 1/32 in.
3/4-1 1/2 in. incl., plus or minus 3/64 in.
1 3/4-2 1/2 in. incl., plus or minus 1/16 in.

DIAMETER OF SHANK — 1/4-1/2 in. incl., size to size minus .005 in.
9/16-1 in. incl., size to size minus .006 in.
1 1/8-2 in. incl., size to size minus .008 in.
2 1/4-2 1/2 in. incl., size to size minus .010 in.

SIZE OF SQUARE — 1/4-1/2 in. incl., size to size minus .004 in.
9/16-1 in. incl., size to size minus .006 in.
1 1/8-2 in. incl., size to size minus .008 in.
2 1/4-2 1/2 in. incl., size to size minus .010 in.

HALF ANGLE — 4 threads per inch, plus or minus 30 min.
4 1/2-5 threads per inch, plus or minus 35 min.
6-9 threads per inch, plus or minus 40 min.
10-28 threads per inch, plus or minus 45 min.

FULL ANGLE — 4 threads per inch, 45 min.
4 1/2-5 threads per inch, 53 min.
6-9 threads per inch, 60 min.
10-28 threads per inch, 68 min.

MAJOR DIAMETER — 1/4-2 1/2 in. incl., maximum = basic + $\sqrt{\frac{\text{basic diam.}}{100}}$
1/4-3/8 in. incl., minimum = maximum minus .0025 in.
7/16-5/8 in. incl., minimum = maximum minus .003 in.
3/4-1 in. incl., minimum = maximum minus .004 in.
1 1/8-1 1/2 in. incl., minimum = maximum minus .0045 in.
1 3/4-2 in. incl., minimum = maximum minus .0055 in.
2 1/4-2 1/2 in. incl., minimum = maximum minus .006 in.

Nut Taps
Ground Thread

Size	Threads per Inch	Pitch Diameter			Major Diameter			General Dimensions				
	NC	Basic	Min.	Max.	Basic	Min.	Max.	A	B	C	D	E
1 3/4	5	1.6201	1.6216	1.6236	1.7500	1.7620	1.7650	14	5 1/2	1 5/8	1.484	1.113
2	4 1/2	1.8557	1.8572	1.8592	2.0000	2.0130	2.0160	15	6 1/8	1 3/4	1.705	1.279
2 1/4	4 1/2	2.1057	2.1072	2.1092	2.2500	2.2630	2.2660	16	6 1/8	1 7/8	1.953	1.465
2 1/2	4	2.3376	2.3396	2.3416	2.5000	2.5140	2.5170	17	6 7/8	2	2.167	1.625

All dimensions are in inches.
All taps have internal center on thread end.

Tolerances

LEAD — plus or minus .0005 in. per inch of thread.

LENGTH OVERALL — 1/4-1 in. incl., plus or minus 1/16 in.
1 1/8-2 1/2 in. incl., plus or minus 3/32 in.

LENGTH OF THREAD — 1/4-1/2 in. incl., plus or minus 1/16 in.
9/16-1 1/2 in. incl., plus or minus 3/32 in.
1 3/4-2 1/2 in. incl., plus or minus 1/8 in.

LENGTH OF SQUARE — 1/4-5/8 in. incl., plus or minus 1/32 in.
3/4-1 1/2 in. incl., plus or minus 3/64 in.
1 3/4-2 1/2 in. incl., plus or minus 1/16 in.

DIAMETER OF SHANK — 1/4-1/2 in. incl., size to size minus .005 in.
9/16-1 in. incl., size to size minus .006 in.
1 1/8-2 in. incl., size to size minus .008 in.
2 1/4-2 1/2 in. incl., size to size minus .010 in.

SIZE OF SQUARE — 1/4-1/2 in. incl., size to size minus .004 in.
9/16-1 in. incl., size to size minus .006 in.
1 1/8-2 in. incl., size to size minus .008 in.
2 1/4-2 1/2 in. incl., size to size minus .010 in.

HALF ANGLE — 4-5 threads per inch, plus or minus 20 min.
6-9 threads per inch, plus or minus 25 min.
10-28 threads per inch, plus or minus 30 min.

MAJOR DIAMETER — 4-5 threads per inch, maximum = basic plus 35 per cent truncation
6-12 threads per inch, maximum = basic plus 40 per cent truncation
13-28 threads per inch, maximum = basic plus 45 per cent truncation
1/4-3/8 in. incl., minimum = maximum minus .0015 in.
3/4-1 in. incl., minimum = maximum minus .002 in.
1 1/8-1 1/2 in. incl., minimum = maximum minus .0025 in.
1 3/4-2 1/2 in. incl., minimum = maximum minus .003 in.

The basic pitch-diameter is the number obtained by subtracting the single-thread depth from the basic major-diameter:

Basic pitch diameter $= D - h$

$D =$ Major diameter
$h =$ Depth of thread
$n =$ Number of threads per inch
$p =$ Pitch of thread

Formulas for Shank Diameters and Sizes of Squares

Machine-Screw Taps

DIAMETER OF SHANK (No. 0 to 5 inclusive) = Maximum major-diameter of No. 6 tap
(No. 6 to 12 inclusive) = Maximum major-diameter to nearest 0.001 in.
SIZE OF SQUARE (No. 0 to 10 inclusive) = Diameter of shank × 0.78 to nearest 0.001 in.
(No. 12) = Diameter of shank × 0.75 to nearest 0.001 in.

Hand Taps

DIAMETER OF SHANK Large shanks = Maximum major-diameter to nearest 0.001 in.
Small shanks = Basic major-diameter minus (Standard V pitch × 1.6) to nearest 0.001 in.
SIZE OF SQUARE = Diameter of shank × 0.75 to nearest 0.001 in.

Taper Taps

DIAMETER OF SHANK (1/4 to 1/2 in. inclusive) = N C basic minor-diameter
(9/16 to 1 in. inclusive) = N C basic minor-diameter minus 0.004 in.
(1⅛ to 2 in. inclusive) = N C basic minor-diameter minus 0.006 in.

Nut Taps

DIAMETER OF SHANK (1/4 to 1/2 in. inclusive) = N C basic minor-diameter
(9/16 to 1 in. inclusive) = N C basic minor-diameter minus 0.004 in.
(1⅛ to 2 in. inclusive) = N C basic minor-diameter minus 0.006 in.
(2⅛ to 2½ in. inclusive) = N C basic minor-diameter minus 0.008 in.
SIZE OF SQUARE = Diameter of shank × 0.75 to nearest 0.001 in.

Pulley Taps

DIAMETER OF SHANK — Maximum major-diameter
SIZE OF SQUARE = Diameter of shank × 0.75 to nearest 0.001 in.

Hand Taps

Cut Thread

Size	Threads per Inch	Pitch Diameter			Major Diameter			General Dimensions				
	NC	Basic	Min.	Max.	Basic	Min.	Max.	A	B	C	D	E
1¾	5	1 6201	1 6216	1 6256	1 7500	1 7575	1 7630	7	3³⁄₁₆	1¼	1 430	1 072
2	4½	1 8557	1 8572	1 8612	2 0000	2 0085	2 0140	7⅝	3⁹⁄₁₆	1⅜	1 644	1 233
2¼	4½	2 1057	2 1072	2 1117	2 2500	2 2590	2 2650	8¼	3⁹⁄₁₆	1⁷⁄₁₆	1 894	1 420
2½	4	2 3376	2 3396	2 3441	2 5000	2 5100	2 5160	8¾	4	1½	2 100	1 575
2¾	4	2 5876	2 5896	2 5946	2 7500	2 7600	2 7670	9¼	4	1⁹⁄₁₆	2 350	1 762
3	3½	2 8144	2 8164	2 8214	3 0000	3 0105	3 0175	9¾	4⁹⁄₁₆	1⅝	2 543	1 907

All dimensions are in inches.
Taps ⁵⁄₁₆ inch and smaller have external center on thread end.
Taps ⅜ inch and larger have internal center.
Cut thread hand taps made according to the above table are regularly furnished with varying number of flutes and length of chamfer. For full information, consult latest trade catalogs.

Tolerances

LEAD	plus or minus .003 in. per inch of thread
LENGTH OVERALL	¼-1 in. incl., plus or minus ¹⁄₃₂ in. 1⅛-3 in. incl., plus or minus ¹⁄₁₆ in.
LENGTH OF THREAD	¼-½ in. incl., plus or minus ¹⁄₁₆ in. ⁹⁄₁₆-1½ in. incl., plus or minus ³⁄₃₂ in. 1¾-3 in. incl., plus or minus ⅛ in.
LENGTH OF SQUARE	¼-1 in. incl., plus or minus ¹⁄₃₂ in. 1⅛-3 in. incl., plus or minus ¹⁄₃₂ in.
DIAMETER OF SHANK	¼-1 in. incl., size to size minus .005 in. 1⅛-2 in. incl., size to size minus .007 in. 2¼-3 in. incl., size to size minus .009 in.
SIZE OF SQUARE	¼-½ in. incl., size to size minus .004 in. ⁹⁄₁₆-1 in. incl., size to size minus .006 in. 1⅛-2 in. incl., size to size minus .008 in. 2¼-3 in. incl., size to size minus .010 in.
HALF ANGLE	3½-4 threads per inch, plus or minus 30 min. 4½-5 threads per inch, plus or minus 35 min. 6-9 threads per inch, plus or minus 40 min. 10-28 threads per inch, plus or minus 45 min.
FULL ANGLE	3½-4 threads per inch, 45 min. 4½-5 threads per inch, 53 min. 6-9 threads per inch, 60 min. 10-28 threads per inch, 68 min.
MAJOR DIAMETER	SIZE ¼-3 in. incl., maximum = basic + $\sqrt{\frac{\text{basic diam.}}{100}}$ ¼-⅜ in. incl., minimum = maximum minus .0025 in. ⁷⁄₁₆-⅝ in. incl., minimum = maximum minus .003 in. ¾-1 in. incl., minimum = maximum minus .004 in. 1⅛-1½ in. incl., minimum = maximum minus .0045 in. 1¾-2 in. incl., minimum = maximum minus .0055 in. 2¼-2½ in. incl., minimum = maximum minus .006 in. 2¾-3 in. incl., minimum = maximum minus .007 in.

Ground Thread

Size	Threads per Inch	Pitch Diameter			Major Diameter			General Dimensions				
	NC	Basic	Min.	Max.	Basic	Min.	Max.	A	B	C	D	E
1¾	5	1 6201	1 6216	1 6236	1 7500	1.7620	1 7650	7	3³⁄₁₆	1¼	1.430	1.072
2	4½	1 8557	1.8572	1.8592	2.0000	2.0130	2 0160	7⅝	3⁹⁄₁₆	1⅜	1.644	1.233
2¼	4½	2.1057	2.1072	2.1092	2.2500	2 2630	2.2660	8¼	3⁹⁄₁₆	1⁷⁄₁₆	1.894	1.420
2½	4	2 2376	2.3396	2 3416	2.5000	2.5140	2.5170	8¾	4	1½	2.100	1.575
2¾	4	2 5876	2 5896	2.5916	2.7500	2.7640	2 7670	9¼	4	1⁹⁄₁₆	2.350	1.762
3	3½	2 8144	2 8164	2 8184	3 0000	3 0150	3.0180	9¾	4⁹⁄₁₆	1⅝	2.543	1.907

All dimensions are in inches. All Taps have internal center on thread end.
Ground thread hand taps made according to the above table are regularly furnished with varying number of flutes and length of chamfer. For full information, consult latest trade catalogs.

Tolerances

LEAD	plus or minus .0005 in. per inch of thread
LENGTH OVERALL	¼-1 in. incl., plus or minus ¹⁄₃₂ in. 1⅛-3 in. incl., plus or minus ¹⁄₁₆ in.
LENGTH OF THREAD	¼-½ in. incl., plus or minus ¹⁄₁₆ in. ⁹⁄₁₆-1½ in. incl., plus or minus ³⁄₃₂ in 1¾-3 in. incl., plus or minus ⅛ in.
LENGTH OF SQUARE	¼-1 in. incl., plus or minus ¹⁄₃₂ in. 1⅛-3 in. incl., plus or minus ¹⁄₁₆ in.
DIAMETER OF SHANK	¼-⅝ in. incl., size to size minus .0015 in. ¾-1½ in. incl., size to size minus .0020 in. 1¾-3 in. incl., size to size minus .0030 in.
SIZE OF SQUARE	¼-½ in. incl., size to size minus .004 in. ⁹⁄₁₆-1 in. incl., size to size minus .006 in. 1⅛-2 in. incl., size to size minus .008 in. 2¼-3 in. incl., size to size minus .010 in.
HALF ANGLE	3½-5 threads per inch, plus or minus 20 min. 6-9 threads per inch, plus or minus 25 min. 10-28 threads per inch, plus or minus 30 min.
MAJOR DIAMETER	3½-5 threads per inch, maximum = basic plus 35 per cent truncation 6-12 threads per inch, maximum = basic plus 40 per cent truncation 13-28 threads per inch, maximum = basic plus 45 per cent truncation ¼-⅝ in. incl., minimum = maximum minus .0015 in. ¾-1 in. incl., minimum = maximum minus .002 in. 1⅛-1½ in. incl., minimum = maximum minus .0025 in. 1¾-3 in. incl., minimum = maximum minus .003 in.

Screw-Threads Division

PERSONNEL

E. H. Ehrman, *Chairman*	Standard Screw Co.
K. L. Herrmann, *Vice-Chairman*	Bantam Ball Bearing Co.
A. Boor	Willys-Overland Co.
E. J. Bryant	Greenfield Tap & Die Corp.
Earle Buckingham	Massachusetts Institute of Technology
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Luther D. Burlingame	Brown & Sharpe Mfg. Co.
George S. Case	Lamson & Sessions Co.
R. M. Heames	Victor Peninsular Division, Allied Products Corp.
J. K. Olsen	Stewart-Warner Speedometer Corp.
D. W. Ovaitt	Buick Motor Co.
O. B. Zimmerman	International Harvester Co.

Wrench Head Bolts and Nuts and Wrench Openings

(*Proposed Revision of American Standard and S.A.E. Standard*)

Since the approval of the original tentative American Standard for the above-mentioned class of products in February, 1927, under the procedure of the American Standards Association, the Sectional Committee was requested by some of the bolt and nut manufacturers to reconsider some of the dimensional details in the report. A questionnaire was accordingly circularized through the Society and the American Society of Mechanical Engineers as sponsors for the Sectional Committee, in April, 1929, which resulted in general approval of the suggested changes. The Sectional Committee accordingly has recommended that they be approved by the sponsors and the American Standards Association and this recommendation was approved at the meeting of members of the Screw Threads Division of the Society's Standards Committee on Dec. 18.

The Division therefore recommends that the Society approve and adopt the accompanying revisions in the American Standard and also the S.A.E. Standard for Cap Screws, Bolts and Nuts and Wrench Openings as follows:

(1) In tables 5, 6, 7 and 10, pp. 318, 319, 320, 321 and 323 of the 1929 edition of the S.A.E. HANDBOOK, change the maximum width across flats of the ⅝-in. size from 15/16 to 1 in. with corresponding corrections for the thickness across flats and also the minimum width across corners for square and hexagon nuts.

(2) In tables 5 and 6, pp. 318, 319 and 321 of the 1929 edition of the S.A.E. HANDBOOK, change the nominal thickness of the 9/16-in. size from 31/64 to ½ in., with corresponding correction of the limits on thickness.

At the time of printing this recommendation of the Division, it was subject to confirmation by Division letter-ballot as a quorum was not present at the Division meeting at which the recommendation was approved.

Slotted-Head Screws

(*Proposed American Standard and S.A.E. Standard*)

Seven years ago the Sectional Committee on Bolt, Nut and Rivet Proportions was organized by the Society and the American Society of Mechanical Engineers as sponsors under the procedure of the American Standards Association to formulate an American Standard for various types of screw and bolt. A sub-committee was organized to prepare a report on Machine Screws, Cap Screws and Wood Screws. This sub-committee, after accumulating and reviewing a large volume of data and holding several meetings, submitted a report in 1924 that was acceptable in the main. Question was raised, however, regarding a number of points in the report and a revised tentative report was circularized in 1928. This report proved to be more acceptable and on submission to the Sectional Committee, was approved and confirmed by 41 of the 47 members of the Sectional Committee, 4 negative votes being cast and 2 members not voting. The report with copies of correspondence bearing on the case were thereupon referred to the Screw-Threads Division of the Society's Standards Committee for ballot as to the acceptance of the report by the Society at the Semi-Annual Meeting last June but the resulting ballot varied so greatly that the subject was withheld for further consideration at a subsequent meeting of the Division.

After a number of delays a meeting of members of the Division was held on Dec. 18 at which the entire report of the Sectional Committee as of April, 1929, was recommended for approval with the exception of Table 2 for Round-Head Machine-Screws. It was further recommended by those present that the head dimensions for Round-Head Machine-Screws, especially sizes No. 8 to ¼ in. inclusive be reconsidered by the Sectional Committee on the basis of criticisms of the report that had been made by automotive users. The recommendations at this meeting of members of the Division were necessarily subject to confirmation by letter-ballot of the entire Division because of the limited attendance at the Division meeting.

The following table shows the differences between the Sectional Committee's proposed machine-screw head-dimensions and those of the General Motors Corporation's standard which is the basis on which it was recommended that the Sectional Committee reconsider its report on Round-Head Machine-Screws. The General Motors standard conforms to the Sectional Committee's report for Wood Screw Heads, the head dimensions of the round-head wood-screws and the round-head machine-screws in the Sectional Committee's report being the same for corresponding screw sizes.

ROUND-HEAD MACHINE-SCREW HEADS[1]

Screw Size	Diameters Maximum	Diameters Minimum	Heights Maximum	Heights Minimum
No. 8	0.309 (0.302)	0.287 (0.280)	0.119 (0.110)	0.107 (0.098)
10	0.359 (0.354)	0.334 (0.329)	0.136 (0.132)	0.124 (0.120)
12	0.408 (0.391)	0.382 (0.375)	0.152 (0.147)	0.140 (0.135)
¼ in.	0.472	0.443	0.174 (0.168)	0.161 (0.155)

[1] Dimensions in ( ) are the General Motors Corp. standard as of February, 1929. All other sizes and dimensions in that standard correspond to the Sectional Committee's report.

The Division therefore submits the accompanying report of the Sectional Committee with the recommendation that it be approved for adoption as American Standard and also S.A.E. Standard with the exception of Table No. 2 for Round-Head Machine-Screws in accordance with the recommendations outlined above.

Flat Head Machine Screws

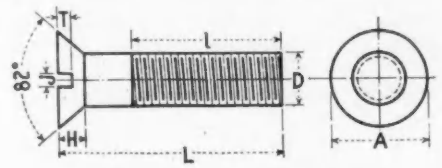


Table No. 1 Head Dimensions

Nominal Size	D	A		H		J		T	
	Max. Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
2	.086	.172	.156	.051	.040	.036	.024	.023	.015
3	.099	.199	.181	.059	.048	.038	.026	.027	.017
4	.112	.225	.207	.067	.055	.040	.028	.030	.020
5	.125	.252	.232	.075	.062	.043	.031	.034	.022
6	.138	.279	.257	.083	.069	.045	.033	.038	.024
8	.164	.332	.308	.100	.084	.050	.037	.045	.029
10	.190	.385	.359	.116	.098	.055	.041	.053	.034
12	.216	.438	.410	.132	.112	.059	.045	.060	.039
1/4	.250	.507	.477	.153	.131	.066	.051	.070	.046
5/16	.3125	.636	.600	.192	.166	.077	.061	.088	.058
3/8	.375	.762	.722	.230	.200	.088	.072	.106	.070

All dimensions in inches.

Note 1. The unthreaded body diameter of machine screws will have approximately the same tolerances as the pitch diameter of the threads shown in the American National Standard for Screw Threads, Class 2 Free Fit, where the number of threads per inch are the same.

Formulas

Head Diameter:
Maximum $A = 2.040\,D - 0.003$
Minimum $A = 1.960\,D - 0.013$

Height of Head:
Maximum $H = 0.619\,D - 0.002$
Minimum $H = 0.552\,D - 0.007$

Width of Slot:
Maximum $J = 0.182\,D + 0.020$
Minimum $J = 0.164\,D + 0.010$

Depth of Slot:
Maximum $T = 0.288\,D - 0.002$
Minimum $T = 0.192\,D - 0.002$

Countersink Angle:
Maximum 82 deg.
Minimum 80 deg.

Round Head Machine Screws

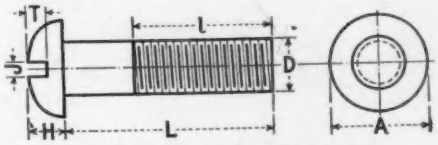


Table No. 2 Head Dimensions

Nominal Size	D	A		H		J		T	
	Max. Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
2	.086	.162	.146	.070	.059	.036	.024	.048	.036
3	.099	.187	.169	.078	.067	.038	.026	.053	.040
4	.112	.211	.193	.086	.075	.040	.028	.058	.043
5	.125	.236	.217	.095	.083	.043	.031	.062	.047
6	.138	.260	.240	.103	.091	.045	.033	.067	.050
8	.164	.309	.287	.119	.107	.050	.037	.076	.057
10	.190	.359	.334	.136	.124	.055	.041	.086	.064
12	.216	.408	.382	.152	.140	.059	.045	.095	.071
1/4	.250	.472	.443	.174	.161	.066	.051	.108	.080
5/16	.3125	.591	.557	.214	.200	.077	.061	.130	.097
3/8	.375	.708	.670	.254	.239	.088	.072	.153	.114

All dimensions in inches.

Note 1. The unthreaded body diameter of machine screws will have approximately the same tolerances as the pitch diameter of the threads shown in the American National Standard for Screw Threads, Class 2 Free Fit, where the number of threads per inch are the same.

Formulas

Head Diameter:
Maximum $A = 1.887\,D$
Minimum $A = 1.813\,D - 0.010$

Height of Head:
Maximum $H = 0.636\,D + 0.015$
Minimum $H = 0.624\,D + 0.005$

Width of Slot:
Maximum $J = 0.182\,D + 0.020$
Minimum $J = 0.164\,D + 0.010$

Depth of Slot:
Maximum $T = 0.362\,D + 0.017$
Minimum $T = 0.268\,D + 0.013$

Shape of Head:—Semi-elliptical.

Oval Head Machine Screws

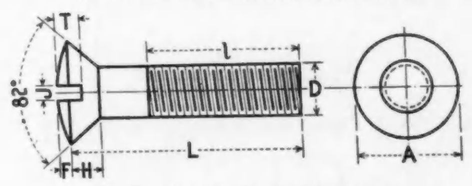


Table No. 3 Head Dimensions

Nominal Size	D	A		H		J		T		F		F+H	
	Max. Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot		Height of Oval		Total Height of Head	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
2	.086	.172	.156	.051	.040	.036	.024	.045	.037	.029	.022	.080	.063
3	.099	.199	.181	.059	.048	.038	.026	.052	.043	.033	.026	.092	.073
4	.112	.225	.207	.067	.055	.040	.028	.059	.049	.037	.029	.104	.084
5	.125	.252	.232	.075	.062	.043	.031	.067	.055	.041	.033	.116	.095
6	.138	.279	.257	.083	.069	.045	.033	.074	.060	.045	.036	.128	.105
8	.164	.332	.308	.100	.084	.050	.037	.088	.072	.053	.043	.152	.126
10	.190	.385	.359	.116	.098	.055	.041	.103	.084	.061	.050	.176	.148
12	.216	.438	.410	.132	.112	.059	.045	.117	.096	.069	.057	.200	.169
1/4	.250	.507	.477	.153	.131	.066	.051	.136	.112	.079	.066	.232	.197
5/16	.3125	.636	.600	.192	.166	.077	.061	.171	.141	.098	.083	.290	.249
3/8	.375	.762	.722	.230	.200	.088	.072	.206	.170	.117	.100	.347	.300

All dimensions in inches.

Note 1. The unthreaded body diameter of machine screws will have approximately the same tolerances as the pitch diameter of the threads shown in the American National Standard for Screw Threads, Class 2 Free Fit, where the number of threads per inch are the same.

Formulas

Head Diameter:
Maximum $A = 2.04\,D - 0.003$
Minimum $A = 1.96\,D - 0.013$

Height of Head:
Maximum $H = 0.619\,D - 0.002$
Minimum $H = 0.552\,D - 0.007$

Countersink Angle:
Maximum 82 deg.
Minimum 80 deg.

Width of Slot:
Maximum $J = 0.182\,D + 0.020$
Minimum $J = 0.164\,D + 0.010$

Depth of Slot:
Maximum $T = 0.556\,D - 0.003$
Minimum $T = 0.460\,D - 0.003$

Height of Oval:
Maximum $F = 0.304\,D + 0.003$
Minimum $F = 0.268\,D - 0.001$

Total Height of Head:
Max. $(F + H)$ = Max. F + Max. H $= 0.923\,D + 0.001$
Min. $(F + H)$ = Min. F + Min. H $= 0.820\,D - 0.008$

Fillister Head Machine Screws

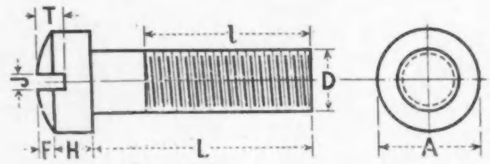


Table No. 4 Head Dimensions

Nominal Size	D	A		H		J		T		F		F+H	
	Max. Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot		Height of Oval		Total Height of Head	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
2	.086	.140	.124	.055	.045	.036	.024	.037	.021	.028	.018	.083	.063
3	.099	.161	.145	.063	.052	.038	.026	.043	.026	.032	.021	.095	.073
4	.112	.183	.166	.072	.060	.040	.028	.048	.031	.035	.024	.107	.084
5	.125	.205	.187	.081	.068	.043	.031	.054	.036	.039	.027	.120	.095
6	.138	.226	.208	.089	.076	.045	.033	.060	.041	.043	.029	.132	.105
8	.164	.270	.250	.106	.091	.050	.037	.071	.050	.050	.035	.156	.126
10	.190	.313	.292	.123	.107	.055	.041	.083	.060	.057	.041	.180	.148
12	.216	.357	.334	.141	.123	.059	.045	.094	.070	.064	.047	.205	.169
¼	.250	.414	.389	.163	.143	.066	.051	.109	.083	.074	.054	.237	.197
5/16	.3125	.519	.490	.205	.181	.077	.061	.137	.106	.092	.068	.297	.249
⅜	.375	.622	.590	.246	.218	.088	.072	.164	.129	.109	.082	.355	.300

All dimensions in inches.

NOTE 1. The unthreaded body diameter of machine screws will have approximately the same tolerance as the pitch diameter of the threads shown in the American National Standard for Screw Threads, Class 2 Free Fit, where the number of threads per inch are the same.

Formulas

Head Diameter:
- Maximum A = 1.670 D — 0.004
- Minimum A = 1.610 D — 0.014

Height of Head (Side):
- Maximum H = 0.660 D — 0.002
- Minimum H = 0.600 D — 0.007

Width of Slot:
- Maximum J = 0.182 D + 0.020
- Minimum J = 0.164 D + 0.010

Depth of Slot:
- Maximum T = 0.440 D — 0.001
- Minimum T = 0.374 D — 0.011

Total Height of Head:
- Max. (F + H) = Max. F + Max. H
- Min. (F + H) = Min. F + Min. H

Height of Oval:
- Maximum F = 0.280 D + 0.004
- Minimum F = 0.220 D — 0.001

Machine Screws

Table No. 5 Preferred Screw Lengths and Heads

Nominal Size	2	3	4	5	6	8	10	12	¼	5/16	⅜	2	3	4	6	8	10	12	¼	5/16	⅜
Length in Inches	COARSE THREAD SERIES, FREE FIT, (Class 2) (Common Pitches) Threads per Inch											FINE THREAD SERIES, FREE FIT, (Class 2) (Less Common Pitches) Threads per Inch									
L	56	48	40	40	32	32	24	24	20	18	16	64	56	48	40	36	32	28	28	24	24
⅛	FRP	FRP	FRP	RP	RP	RP						FR	FRP	FRP	RP						
3/16	FROP	FROP	FROP	FRP	FROP	FRP	RP					FR	FRP	FRP	RP	RP	R				
¼	FROP	FROP	FROP	FRP	FROP	FROP	FROP	RP				FR	FRP	FRP	FRP	FRP	FROP				
5/16	FRP	FRP	FROP	FRP	FROP	FROP	FROP	FRP	FRP			FR	FR	FRP	FRP	FRP	FROP	F	F		
⅜	FRP	FRP	FROP	FRP	FROP	FROP	FROP	FROP	FRP			F	FR	FRP	FROP	FROP	FROP	FROP	FR		
7/16	FRP	FRP	FROP	FRP	FROP	FROP	FROP	FROP	FRP			F	FR	FRP	FROP	FROP	FROP	FROP	FR		
½	FRP	FRP	FROP	FRP	FROP	FROP	FROP	FROP	FROP	FROP	FRP	F	FR	FRP	FROP	FROP	FROP	FROP	FR	R	
⅝	FRP	FRP	FROP	FRP	FROP	FROP	FROP	FROP	FROP	FROP	FRP		R	FR	FRP	FROP	FROP	FROP	FRP	FRP	FR
¾	FR	FR	FRP	FRP	FROP	FROP	FROP	FROP	FROP	FROP	FROP		R	FR	FRP	FROP	FROP	FROP	FRP	FRP	FR
⅞	R	R	FRP	FRP	FROP	FROP	FROP	FROP	FROP	FROP	FROP				FROP	FROP	FROP	FRP	FRP	FRP	FR
1			FRP	FRP	FROP	FROP	FROP	FROP	FROP	FRP	FRP				FR	FROP	FROP	FRP	FRP	FR	FR
1¼			R	FR	FROP	FROP	FROP	FROP	FROP	FRP	FRP				FR	FRP	FROP	FRP	FRP	FR	F
1½			R	FR	FROP	FROP	FROP	FROP	FROP	FRP	FRP				R	FR	FROP	FRP	FRP	F	F
1¾				F	FROP	FROP	FROP	FROP	FROP	FRP	FRP					F	FROP	FRP	FRP	F	F
2					FROP	FROP	FROP	FROP	FROP	FRP	FRP					F	FROP	FROP	FRP	F	F
2¼					R	FRP	FRP	FRP	FROP	FRP	FRP					F	FRP	FR	FR	F	F
2½						FR	FRP	FRP	FRP	FRP	FRP					F	FRP	F	F	F	F
2¾						FR	FR	FR	FRP	FRP	FRP					F	FRP	F	F	F	F
3						FR	FR	FR	FRP	FRP	FRP					F	FRP	F	F	F	F

Note 1. The table of screw lengths reproduced above is intended only as a guide to the users of these screws. A number of the listed sizes and lengths will not be regularly stocked by the manufacturers but will be available on order of a sufficient quantity. Letters in the vertical column under the nominal screw sizes indicate the style of head for a particular length of screw; hence, F = Flat Head, R = Round Head, O = Oval Head, P = Fillister Head.

Note 2. Tolerance in length (L) = — 3 per cent; but not less than — .025 inches.

Note 3. Where length of screw (L) is 1¼ inches or less, the length of thread (*l*) will extend to as near the head as is practicable. Where the length of screw is over 1¼ inches the thread length shall be not less than 1¼ inches.

Flat Head Cap Screws

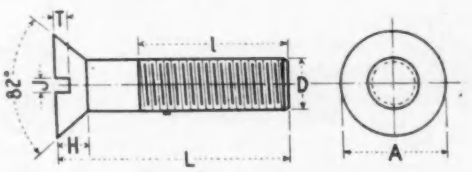


Table No. 6 Head Dimensions

Nominal Size	D	A		H	J		T	
	Max. Diameter	Head Diameter		Height of Head	Width of Slot		Depth of Slot	
		Maximum	Minimum	Nominal	Maximum	Minimum	Maximum	Minimum
1/4	250	1/2	.477	.146	.070	.058	.073	.053
5/16	3125	5/8	.598	.183	.079	.065	.091	.066
3/8	375	3/4	.719	.220	.088	.074	.110	.080
7/16	4375	13/16	.780	.220	.098	.083	.110	.075
1/2	500	7/8	.841	.220	110	.094	.110	.070
9/16	5625	1	.962	.256	.123	.106	.128	.083
5/8	625	1 1/8	1 083	.293	.138	.119	.146	.096
3/4	750	1 3/8	1 326	.366	.154	.134	.183	.123

All dimensions in inches.

Note 1 The unthreaded body diameter of Cap Screws will have the same tolerances as the major thread diameter shown in the American National Standard for Screw Threads, Class 2 Free Fit, where the number of threads per inch are the same.

Formulas

Head Diameter:
Maximum A = Nominal (No formula)
Minimum A = Maximum A — (.03 Max. A + .008)
Height of Head:
Nominal H = Computed from Maximum D and Maximum A, with 81 deg. (mean) angle. Nominal H, subject to variations in A and D and angle.
Width of Slot:
Nominal J = .160 D + .024 = S (Approx. Cutter Width)
Maximum J = S + (.05 S + .003)
Minimum J = S — (.05 S + .003)
Depth of Slot:
Maximum T = 0.500 H Nominal
Minimum T = Maximum T — .08 D
Countersink Angle:
Maximum 82 deg.
Minimum 80 deg.

Button Head Cap Screws

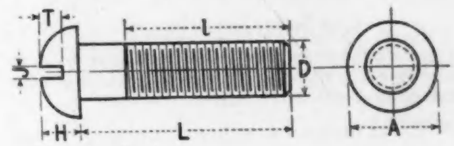


Table No. 7 Head Dimensions

Nominal Size	D	A		H		J		T	
	Max. Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
1/4	.250	7/16	.418	.191	.175	.070	.058	.117	.097
5/16	.3125	9/16	.541	.246	.226	.079	.065	.151	.126
3/8	.375	5/8	.602	.273	.251	.088	.074	.167	.137
7/16	.4375	3/4	.725	.328	.302	.098	.083	.202	.167
1/2	.500	13/16	.786	.355	.328	.110	.094	.219	.179
9/16	.5625	15/16	.908	.410	.379	.123	.106	.253	.208
5/8	.625	1	.970	.438	.405	.138	.119	.270	.220
3/4	.750	1 1/4	1.215	.547	.506	.154	.134	.337	.277

All dimensions in inches.

Note 1. The unthreaded body diameter of Cap Screws will have the same tolerances as the major thread diameter shown in the American National Standard for Screw Threads, Class 2 Free Fit, where the number of threads per inch are the same.

Formulas

Head Diameter:
Maximum A = Nominal (No formula)
Minimum A = Maximum A — (.02 Max. A + .010)
Height of Head:
Maximum H = 7/16 Maximum A
Minimum H = Maximum H — (0.03 Max. A + .003)
Width of Slot:
Nominal J = .160 D + .024 = S (Approx. Cutter Width)
Maximum J = S + (.05 S + .003)
Minimum J = S — (.05 S + .003)
Depth of Slot:
Maximum T = 2/3 Minimum H
Minimum T = Maximum T — .08 D
Shape of Head: Semi-elliptical.

Fillister Head Cap Screws

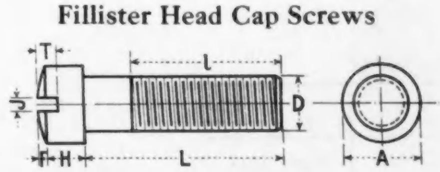


Table No. 8 Head Dimensions

Nominal Size	D	A		H		J		T		F		F+H	
	Max. Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot		Height of Oval		Total Height of Head	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
1/4	250	3/8	363	11/64	.157	.070	.058	.097	.077	.044	.038	.216	.195
5/16	3125	7/16	424	13/64	.186	.079	.065	.115	.090	.050	.044	.253	.230
3/8	375	9/16	.547	1/4	.229	.088	.074	.142	.112	.064	.056	.314	.285
7/16	4375	5/8	608	19/64	.274	.098	.083	.168	.133	.071	063	.368	.337
1/2	500	3/4	731	21/64	.301	.110	.094	.188	.148	.084	.075	.412	.376
9/16	5625	13/16	.792	3/8	.347	.123	.106	.214	.169	.091	081	.466	.428
5/8	625	7/8	853	27/64	.392	.138	.119	.240	.190	.099	088	.521	.480
3/4	750	1	976	1/2	.466	.154	.134	.283	.233	.112	.100	.612	.566
7/8	875	1 1/8	1 098	19/32	.556	.173	.151	.334	.264	.126	.113	.720	.669
1	1 000	1 5/16	1 282	21/32	613	.194	.170	.372	.292	.146	.131	.802	.744

All dimensions in inches.

Note 1. The unthreaded body diameter of Cap Screws will have the same tolerances as the mojor thread diameter shown shown in the American National Standard for Screw Threads, Class 2 Free Fit, where the number of threads per inch are the same.

Formulas

Head Diameter:
Maximum A = Nominal (No formula)
Minimum A = Maximum A — (.02 Max. A + .004)
Height of Head (Side):
Maximum H = 2/3 D to nearest 64th inch
Minimum H = Maximum H — (0.03 Max. A + .004)
Width of Slot:
Nominal J = .160 D + .024 = S (Approx. Cutter Width)
Maximum J = S + (.05 S + .003)
Minimum J = S — (.05 S + .003)

Depth of Slot:
Maximum T = 0.50 (Minimum F + Minimum H)
Minimum T = Maximum T — .08 D
Height of Oval:
Maximum F = 0.110 Maximum A + .002
Minimum F = 0.100 Maximum A
Total Height of Head:
Maximum (F + H) = Max. F + Max. H
Minimum (F + H) = Min. F + Min. H

Formulas for F included for purpose of determining tolerances for T and F, and are not to be used for inspection purposes.

Cap Screw Lengths

The length shall be measured from the largest diameter of the bearing surface of the Head to the extreme point, n a line parallel to the axis of the screw.

The difference between consecutive lengths of screws:

For screw lengths ¼ in. to 1 in. shall be ⅛ in.
" " " 1 in. to 4 in. " " ¼ in.
" " " 4 in. to 6 in. " " ½ in.

The tolerance in screw length shall be 3 per cent of the nominal length, with a minimum tolerance of .030 inch; one-third of the tolerance to be applied minus, and two-thirds plus.

Thread Lengths

Slotted Head Cap Screws shall be regularly threaded coarse pitch, and when so threaded shall have a length of thread equal to 2 D + ¼ inch. Screws too short to allow the formula length of thread, may be threaded as close to the Head as practicable.

Screw Points

The points of all Cap Screws shall be flat, the flat being normal to the axis of the screw; and shall be chamfered at an angle of 35 deg. with the surface of the flat, + 5 deg., –0 deg.; the chamfer to extend to the bottom of the thread. The edge of the chamfer is to be slightly rounded.

Flat Head Wood Screws

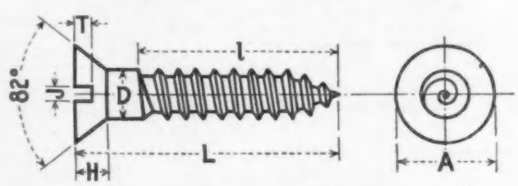


Table No. 10 Head Dimensions

Screw Number	D	A		H		J		T	
	Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
0	.060	.119	.105	.035	.026	.031	.020	.015	.010
1	.073	.146	.130	.043	.033	.033	.022	.019	.012
2	.086	.172	.156	.051	.040	.036	.024	.023	.015
3	.099	.199	.181	.059	.048	.038	.026	.027	.017
4	.112	.225	.207	.067	.055	.040	.028	.030	.020
5	.125	.252	.232	.075	.062	.043	.031	.034	.022
6	.138	.279	.257	.083	.069	.045	.033	.038	.024
7	.151	.305	.283	.091	.076	.047	.035	.041	.027
8	.164	.332	.308	.100	.084	.050	.037	.045	.029
9	.177	.358	.334	.108	.091	.052	.039	.049	.032
10	.190	.385	.359	.116	.098	.055	.041	.053	.034
11	.203	.411	.385	.124	.105	.057	.043	.056	.037
12	.216	.438	.410	.132	.112	.059	.045	.060	.039
14	.242	.491	.461	.148	.127	.064	.050	.068	.044
16	.268	.544	.512	.164	.141	.069	.054	.075	.049
18	.294	.597	.563	.180	.155	.074	.058	.083	.054
20	.320	.650	.614	.196	.170	.078	.062	.090	.059
24	.372	.756	.716	.228	.198	.088	.071	.105	.069

All dimensions in inches.

Tolerance in diameter (D) + 0.004 to — 0.007 inches

Formulas

Head Diameter:
Maximum A = 2.04 D — 0.003
Minimum A = 1.96 D — 0.013
Height of Head:
Maximum H = 0.619 D — 0.002
Minimum H = 0.552 D — 0.007
Width of Slot:
Maximum J = 0.182 D + 0.020
Minimum J = 0.164 D + 0.010
Depth of Slot:
Maximum T = 0.288 D — 0.002
Minimum T = 0.192 D — 0.002
Countersink Angle:
Maximum 82 deg.
Minimum 80 deg.

Round Head Wood Screws

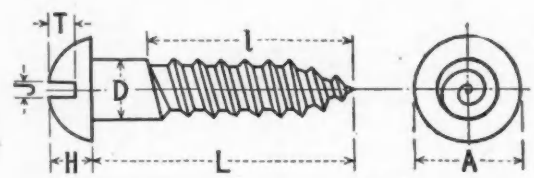


Table No. 9 Head Dimensions

Screw Number	D	A		H		J		T	
	Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
0	.060	.113	.099	.053	.042	.031	.020	.039	.029
1	.073	.138	.122	.061	.051	.033	.022	.043	.033
2	.086	.162	.146	.070	.059	.036	.024	.048	.036
3	.099	.187	.169	.078	.067	.038	.026	.053	.040
4	.112	.211	.193	.086	.075	.040	.028	.058	.043
5	.125	.236	.217	.095	.083	.043	.031	.062	.047
6	.138	.260	.240	.103	.091	.045	.033	.067	.050
7	.151	.285	.264	.111	.099	.047	.035	.072	.053
8	.164	.309	.287	.119	.107	.050	.037	.076	.057
9	.177	.334	.311	.128	.115	.052	.039	.081	.060
10	.190	.359	.334	.136	.124	.055	.041	.086	.064
11	.203	.383	.358	.144	.132	.057	.043	.090	.067
12	.216	.408	.382	.152	.140	.059	.045	.095	.071
14	.242	.457	.429	.169	.156	.064	.050	.105	.078
16	.268	.506	.476	.185	.172	.069	.054	.114	.085
18	.294	.555	.523	.202	.188	.074	.058	.123	.092
20	.320	.604	.570	.219	.205	.078	.062	.133	.099
24	.372	.702	.664	.252	.237	.088	.071	.152	.113

All dimensions in inches.

Tolerance in diameter (D) = + 0.004 to — 0.007 inches.

Formulas

Head Diameter:
Maximum A = 1.887 D
Minimum A = 1.813 D — 0.010
Height of Head:
Maximum H = 0.636 D + 0.015
Minimum H = 0.624 D + 0.005
Width of Slot:
Maximum J = 0.182 D + 0.020
Minimum J = 0.164 D + 0.010
Depth of Slot:
Maximum T = 0.362 D + 0.017
Minimum T = 0.268 D + 0.013
Shape of Head: Semi-elliptical.

Oval Head Wood Screws

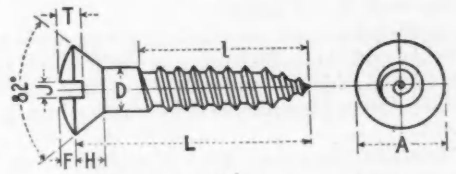


Table No. 11 Head Dimensions

Screw Number	D	A		H		J		T		F		F+H	
	Diameter	Head Diameter		Height of Head		Width of Slot		Depth of Slot		Height of Oval		Total Height of Head	
		Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
0	.060	119	.105	035	026	.031	020	.030	.025	.021	.015	056	.041
1	.073	.146	130	043	033	.033	022	.038	.031	.025	.019	068	.052
2	.086	.172	.156	051	040	.036	024	.045	.037	.029	.022	.080	.063
3	.099	.199	.181	.059	.048	.038	026	.052	.043	.033	.026	092	.073
4	.112	.225	.207	067	.055	.040	028	.059	.049	.037	.029	104	.084
5	.125	.252	.232	.075	062	.043	031	.067	.055	.041	.033	.116	.095
6	.138	.279	.257	.083	.069	.045	033	.074	.060	.045	.036	.128	.105
7	.151	.305	.283	.091	.076	.047	035	.081	.066	.049	039	.140	.116
8	.164	.332	.308	.100	.084	.050	037	.088	.072	.053	043	.152	.126
9	.177	.358	.334	.108	.091	.052	.039	.095	.078	.057	046	.164	.137
10	.190	.385	.359	.116	.098	.055	.041	.103	.084	.061	050	.176	.148
11	.203	.411	.385	.124	.105	.057	.043	.110	.090	.065	053	.188	.158
12	.216	.438	.410	.132	.112	.059	.045	.117	.096	.069	057	.200	.169
14	.242	.491	.461	.148	.127	.064	.050	.132	.108	.077	.064	224	.190
16	.268	.544	.512	.164	.141	.069	.054	.146	120	.084	071	248	.212
18	.294	.597	.563	.180	.155	.074	058	.160	.132	092	078	272	.233
20	.320	.650	.614	.196	.170	078	062	.175	.144	100	085	.296	.254
24	.372	.756	.716	.228	.198	088	.071	.204	.168	.116	.099	344	297

All dimensions in inches.

Tolerance in diameter (D) = + 0.004 to — 0.007 inches

Formulas

Head Diameter:
Maximum A = 2.04 D — 0.003
Minimum A = 1.96 D — 0.013
Height of Head (Countersink):
Maximum H = 0.619 D — 0.002
Minimum H = 0.552 D — 0.007
Width of Slot:
Maximum J = 0.182 D + 0.020
Minimum J = 0.164 D + 0.010
Depth of Slot:
Maximum T = 0.556 D — 0.003
Minimum T = 0.460 D — 0.003
Height of Oval:
Maximum F = 0.304 D + 0.003
Minimum F = 0.268 D — 0.001
Total Height of Head:
Max. (F + H) = Max. F + Max. H = 0.923 D + 0.001
Min. (F + H) = Min. F + Min. H = 0.820 D — 0.008
Countersink Angle:
Maximum 82 deg.
Minimum 80 deg.

Table No. 12 Length Tolerances of Round Head Wood Screws

Screw Number	Tolerance Minus
0	06
1	07
2	08
3	08
4	.09
5	.10
6	.10
7	11
8	12
9	13
10	13
11	.14
12	.15
14	.16
16	.18
18	.20
20	.22
24	.27

Tolerances in inches.

Table No. 13 Lengths of Flat and Oval Head Wood Screws

Nominal	Max.	Min.
¼	250	.22
⅜	.375	.34
½	500	46
⅝	625	.59
¾	750	.71
⅞	.875	83
1	1 000	96
1 ¼	1 250	1 20
1 ½	1.500	1.45
1 ¾	1.750	1 69
2	2.000	1 94
2 ¼	2 250	2 19
2 ½	2.500	2.43
2 ¾	2.750	2 68
3	3 000	2 92
3 ½	3.500	3.42
4	4 000	3 91
4 ½	4.500	4.40
5	5.000	4.89

All dimensions in inches.

Brass Wood Screws

Table No. 14 Standard Screw Lengths and Heads

Screw Number / Nominal Length	0	1	2	3	4	5	6	7
¼	FRO	FRO	FRO	FRO	FRO			
⅜	FRO	FRO	FRO	FRO	FRO	FRO	FRO	
½		FRO	FRO	FRO	FRO	FRO	FRO	FRO
⅝			FRO	FRO	FRO	FRO	FRO	FRO
¾			FRO	FRO	FRO	FRO	FRO	FRO
⅞					FRO	FRO	FRO	FRO
1					FRO	FRO	FRO	FRO
1¼							FRO	FRO
1½							FRO	FRO
1¾								
2								

Screw Number / Nominal Length	8	9	10	11	12	14	16	18
½	FRO							
⅝	FRO	FRO	FRO					
¾	FRO	FRO	FRO	FR	FRO			
⅞	FRO	FRO	FRO	FR	FR			
1	FRO	FRO	FRO	FR	FRO	FRO		
1¼	FRO	FRO	FRO	FR	FRO	FRO		
1½	FRO	FRO	FRO	FR	FRO	FRO		
1¾	FRO	FRO	FRO	FR	FRO	FRO		
2	FRO	FRO	FRO	FR	FRO	FRO	F	F
2¼			FR	FR	FR	FR	F	F
2½			FR	FR	FR	FR	F	F
2¾								
3					FR	FR	F	F
3½					F	F	F	F

All dimensions in inches.

Note 1. Letters in the vertical columns under screw numbers indicate the style of head for that particular length; hence, F = Flat Head, R = Round Head, O = Oval Head.

Steel Wood Screws

Table No. 15 Standard Screw Lengths and Heads

Screw Number / Nominal Length	0	1	2	3	4	5	6	7	8
¼	FR	FR	FR	FR	FR				
⅜	FR	FR	FR	FRO	FRO	FR	FR	FR	FR
½		FR	FR	FR	FRO	FRO	FRO	FRO	FRO
⅝			FR	FR	FRO	FRO	FRO	FRO	FRO
¾			FR	FR	FRO	FRO	FRO	FRO	FRO
⅞				FR	FR	FR	FR	FRO	FRO
1				FR	FR	FRO	FRO	FRO	FRO
1¼					FR	FR	FRO	FRO	FRO
1½					FR	FR	FRO	FRO	FRO
1¾							FRO	FRO	FRO
2							FR	FR	FR
2¼							FR	FR	FR
2½							FR	FR	FR
2¾									FR

Screw Number / Nominal Length	9	10	11	12	14	16	18	20	24
½	FR	FR							
⅝	FR	FR	FR	FR					
¾	FRO	FRO	FR	FR					
⅞	FRO	FRO	FR	FRO	FR				
1	FRO	FRO	FRO	FRO	FRO	FR			
1¼	FRO	FRO	FRO	FRO	FRO	FR	FR		
1½	FRO	FRO	FRO	FRO	FRO	FRO	FRO	FRO	
1¾	FRO	FRO	FRO	FRO	FRO	FRO	FRO	FRO	
2	FR	FRO	FRO	FRO	FRO	FRO	FRO	FRO	
2¼	FR	FR	FR	FRO	FRO	FR	F	F	
2½	FR	FR	FR	FRO	FRO	FR	FR	FR	
2¾	FR	FR	F	FR	FR	F	F	F	
3	FR	FR	F	FR	FR	FR	FR	F	F
3½		FR	F	FR	FR	FR	F	F	F
4				FR	FR	FR	F	F	F
4½					F	F	F	F	F
5					F	F	F	F	F

All dimensions in inches.

Note 1. Letters in the vertical columns under screw numbers indicate the style of head for that particular length, hence, F = Flat Head, R = Round Head, O = Oval Head.

Tire and Rim Division

PERSONNEL

H. M. Crane, *Chairman*	General Motors Corp.
T. J. Litle, Jr., *Vice-Chairman*	Marmon Motor Car Co.
C. S. Ash	Wire Wheel Corp. of America
R. S. Begg	Stutz Motor Car Co.
C. Breer	Chrysler Sales Corp.
C. C. Carlton	Motor Wheel Corp.
L. A. Chaminade	Studebaker Corp.
B. Darrow	Goodyear Tire & Rubber Co.
E. E. Dearth	Fisk Rubber Co.
W. T. Fishleigh	Ford Motor Co.
A. G. Geistert	Adam Opel A. G.
T. G. Graham	B. F. Goodrich Co.
W. R. Griswold	Packard Motor Car Co
J. E. Hale	Firestone Tire & Rubber Co.
H. W. Kranz	Cleveland Welding Co.
B. J. Lemon	United States Rubber Co.
E. S. Marks	H. H. Franklin Mfg. Co.
W. B. Minch	General Motors Corp.
Maurice Olley	Rolls-Royce of America, Inc.
G. E. Parker	Cadillac Motor Car Co.
A. J. Scaife	White Co.
J. G. Swain	Firestone Steel Products Co.
C. P. Thomas	Reo Motor Car Co.
L. Thoms	Graham-Paige Motors Corp.
F. E. Watts	Hupp Motor Car Corp.

Tires and Rims for Passenger Cars

(Proposed Revision of S.A.E. Standard)

Since the last revision of the table of passenger-car tires and rims for original equipment, considerable changes have taken place in tire requirements necessitating a further revision at this time.

While the number of sizes included in the present table provided 19 different tires to meet requirements at the time the table was approved, reducing the number of tires for original equipment further to 15 and at the same time providing a suitable series of sizes for all engineering purposes has been found possible.

The table submitted herewith for approval provides for the inclusion of the 5.25-18 tire and the elimination of the 4.50-20, 4.50-21, 4.75-20 and 5.00-20 in view of the fact that these tires have been or will be discontinued as original equipment.

The Division requests that the Standards Committee approve this table as revised.

Balloon Tires and Rims for Commercial Vehicles

(Proposed Revision of S.A.E. Standard)

Herewith is given a table on balloon tires and rims for commercial vehicles representing the recommended revision of the S.A.E. Standard printed on p. 68, of the Supplement to the 1929 edition of the S.A.E. HANDBOOK. The 10.50-20 tire on the 8-in. rim has been eliminated and the dimension for the maximum tire-width on the rim for the 10.50-20 tire on the 9-10-in. rim has been changed from 10.80 to 10.95. Four new tire sizes have been added; 5.00-20, 5.50-20, 6.00-20, 6.50-20, and the flange heights have been revised in accordance with the latest practice of the Tire and Rim Association as approved at its meeting in August. This table is submitted for approval as revised.

High-Pressure Tires and Rims for Commercial Vehicles

(Proposed Revision of S.A.E. Standard)

The Standards Committee is requested to approve the revision of the present specification on p. 69 of the Supplement to the 1929 edition of the S.A.E. HANDBOOK in accordance with the table given herewith, wherein the flange heights have been revised in accordance with the recommendations of the Tire and Rim Association for rim dimensions.

ORIGINAL-EQUIPMENT TIRES AND RIMS FOR PASSENGER CARS

NOT "SUPER" TIRES

(Proposed Revision of S.A.E. Standard)

The accompanying table of passenger-car tire and rim sizes and dimensions was adopted by the Society as representing those that meet controlling engineering requirements and being suitable for original equipment on passenger cars. The sizes shown include those actually used sufficiently to warrant including them in the table as standard. The dimensions of the tires and rims correspond with those adopted by the Tire and Rim Association of America as of Dec. 13, 1929.

	Tire Sections, In.						
Nominal Rim-Diameter, In.	4.75	5.00	5.25	5.50	6.00	6.50	7.00
18	....	....	5.25–18	5.50–18	6.00–18	6.50–18	7.00–18
19	4.75–19	5.00–19	5.25–19	5.50–19	6.00–19	6.50–19	7.00–19
20	....	....	....	....	6.00–20	6.50–20	7.00–20
Flat-Base Rim-Section, Width, In. Nominal	....	4	4	4	4½	4½	5
Actual	2.75-*D*	2.69-*E*	2.69-*E*	2.69-*E*	3.12-*F*	3.12-*F*	3.75-*F*
Maximum Tire-Width on Flat-Base Rim, In.	5.05	5.15	5.35	5.60	5.95	6.40	6.90
Drop-Center Rim-Section Width, In.	3.00-*D*	3.00-*D*	3.25-*E*	3.25-*E*	3.62-*F*	3.62-*F*	4.00-*F*
Maximum Tire-Width on Drop-Center Rim, In.	5.20	....	....	....	....	....	7.05

Method of Marking for Balance.—Tires shall be marked for balance by a red dot or a ¼-in. square on the serial side of the casing just above the rim flange.

BALLOON TIRES AND RIMS FOR COMMERCIAL VEHICLES

(Proposed Revision of S.A.E. Standard)

The accompanying table of commercial-vehicle tire and rim sizes and dimensions was adopted by the Society as representing those that meet controlling engineering requirements and being suitable for original equipment on commercial vehicles. The sizes shown include those actually used sufficiently to warrant including them in the table as standard. The dimensions of the tires and rims correspond with those adopted by the Tire and Rim Association of America as of Dec. 13, 1929.

	Tire Sections, In.								
Nominal Rim-Diameter, In.	5.50	6.00	6.50	7.00	7.50	8.25	9.00	9.75	10.50
20	5.50–20	6.00–20	6.50–20	7.00–20	7.50–20	8.25–20	9.00–20	9.75–20	10.50–20
22	....	....	....	....	....	8.25–22	9.00–22	9.75–22	....
Nominal Rim-Section Width, In.	5	5	5	5	6	7	7	8	9–10
Actual Rim-Width between Flanges, In.	3.75	3.75	3.75	3.75	4.33	5.00	5.00	6.00	7.33
Rim-Flange Height, In.	1	1	1	1	1 1/8	1 5/16	1 5/16	1 1/2	1 3/4
Maximum Permissible Tire-Width, In.	....	6.20	6.70	7.20	7.75	8.50	9.25	10.05	10.95

HIGH-PRESSURE TIRES AND RIMS FOR COMMERCIAL VEHICLES

(Proposed Revision of S.A.E. Standard)

The accompanying table of commercial-vehicle tire and rim sizes and dimensions was adopted by the Society as representing those that meet controlling engineering requirements and being suitable for original equipment on commercial vehicles. The sizes shown include those actually used sufficiently to warrant including them in the table as standard. The dimensions of the tires and rims correspond with those adopted by the Tire and Rim Association of America as of Dec. 13, 1929.

	Tire Sections, In.						
Nominal Rim-Diameter, In.	5	6[a]	7	8	9	9[b]	10
20	30x5	32x6	34x7	36x8	38x9	38x9	40x10
24	34x5	36x6	38x7	40x8	42x9	42x9	44x10
Rim-Section, Width, In. Nominal	5	6	7	8	8	9–10	9–10
Actual	3.75	4.33	5.00	6.00	6.00	7.33	7.33
Rim-Flange Height, In.	1	1 1/8	1 5/16	1 1/2	1 1/2	1 3/4	1 3/4
Maximum Permissible Tire-Width on Rim, In.	6.05	6.90	8.05	9.20	10.15	10.60	11.50

[a] Eight-ply 6 in.—20 to be 6.55 in. maximum on 5-in. truck rim.
[b] Recommended practice.

Passenger-Car Tire-Load and Inflation-Pressure Table, Truck and Bus Balloon-Tire Load and Inflation-Pressure Table, Dual Spacing for Truck and Motorcoach High-Pressure and Balloon Tires.

(Proposed Revision of S.A.E. Recommended Practices)

The Division has approved the revision of the above-listed specifications, which were printed on pp. 70, 72, 73, 74 and 75 respectively, of the Supplement to the 1929 edition of the S.A.E. HANDBOOK, in accordance with the latest recommendations of the Tire and Rim Association, and submits the revised tables herewith to the Standards Committee for approval as revised.

Dual Spacings for Truck and Motorcoach Balloon and High Pressure Tires

(Proposed Revision of S.A.E. Recommended Practice)

The accompanying table is the standard of the Tire and Rim Association of America, as of its meeting of Dec. 13, 1929, and is published by the Society as information supplementing the tables of S.A.E. Standard tire and rim sizes and dimensions adopted by the Society in January, 1930.

DUAL-SPACING DIMENSIONS

Rim Size, In.	Spacing, Center to Center Tire and Rim, In.	Tire Size, In. Permits Oversizing	Tire Size, In. Does Not Permit Oversizing
5	7 1/4	5.00	5.00
5	7 3/4	5.00	6.00[c]
5	7 1/4	5.50	6.00
5	7 3/4	6.00	6.50
5	8 1/4	6.50	7.00
6	9	6.00	
6	9	7.00	7.50
7	10	7.00	
7	10	7.50	8.25
7	10 1/2	8.25	9.00
8	11 1/2	8.00	
8	11 1/2	9.00	9.75
8	12	9.75	10.50
9-10	12 3/4	9.00	10.00
9-10	12 3/4		10.50

[c] Eight-ply only.

PASSENGER-CAR TIRE-LOAD AND INFLATION-PRESSURE TABLE

(Proposed Revision of S.A.E. Recommended Practice)

The accompanyirg table is the standard of the Tire and Rim Association of America, as of its meeting of Dec. 13, 1929, and is published by the Society as information supplementing the tables of S.A.E. Standard tire and rim sizes and dimensions adopted by the Society in January, 1930.

Minimum Inflation-Pressure Lb.	Tire Sections and Rim Diameters, In.																		Minimum Inflation-Pressure Lb.
	4.40		4.50		4.75		5.00		5.25		5.50		6.00		(6.20) 6.50		(6.75) 7.00		
	18 and 19	20 and 21	18 and 19	20 and 21	18 and 19	20 and 21	18 and 19	20 and 21	18 and 19	20 and 21	18 and 19	20 and 21	18 and 19	20 and 21	18 and 19	20 and 21	18 and 19	20 and 21	
	Maximum Load Per Tire, Lb.																		
28	610	650	650	700	700	745	745	815	815	880	880	925	950	1,025	1,075	1,140	1,200	1,300	28
30	660	700	700	750	750	800	800	870	870	940	940	1,000	1,025	1,105	1,155	1,230	1,300	1,400	30
32	710	750	750	800	800	855	855	925	925	1,000	1,000	1,075	1,100	1,190	1,240	1,320	1,400	1,500	32
34	760	800	800	850	850	910	910	980	980	1,060	1,060	1,150	1,175	1,270	1,320	1,410	1,500	1,600	34
36	810	850	850	900	900	965	965	1,035	1,035	1,120	1,120	1,225	1,250	1,350	1,400	1,500	1,600	1,700	36

Single underscoring denotes maximum recommended-loads for 4-ply tires.

Double underscoring denotes maximum recommended-loads for 6-ply tires.

Note 1.—In the above table the weights are based on average passenger load using 150 lb. as weight per passenger and calculated as follows:

All two, three, and four-passenger cars, two-passenger load (midway).
All five-passenger cars, three-passenger load, two midway and one over rear axle.
All seven-passenger cars, four-passenger load, two midway and two over rear axle.

Note 2.—This table not to be used for busses or trucks.

Note 3.—To take care of front-wheel steering and tread-wear conditions, it is recommended that front tires be inflated to same or greater pressure as rear tires, depending on recommendations of car manufacturers.

TRUCK AND BUS BALLOON TIRE-LOAD AND INFLATION-PRESSURE TABLE

(Proposed Revision of S.A.E. Recommended Practice)

The accompanying table is the standard of the Tire and Rim Association of America, as of its meeting of Dec. 13, 1929, and is published by the Society as information supplementing the tables of S.A.E. Standard tire and rim sizes and dimensions adopted by the Society in January, 1930.

Minimum Inflation-Pressure, Lb.	Tire Sections and Rim Diameters, In.								
	5.50	6.00	6.50	7.00	7.50	8.25	9.00	9.75	10.50
35	1,075								
40	1,225	1,250							
45		1,400	1,450	1,700					
50			1,650	1,900	1,975	2,250			
55					2,200	2,525	2,900		
60						2,800	3,200	3,600	
65							3,500	3,900	4,400
70								4,200	4,700
75									5,000

Underscoring indicates maximum recommended-loads.

S. A. E. Journal

MAY, 1930

REPORTS OF DIVISIONS TO STANDARDS COMMITTEE

The Society of Automotive Engineers, Inc.
29 West Thirty-ninth Street
New York City

Reports of Divisions to Standards Committee

Standards Committee Meeting May 25

French Lick Springs Hotel—French Lick, Ind.

IN this pamphlet are printed reports that have been prepared for submission to the Standards Committee and to the Society by Divisions of the Standards Committee since the Annual Meeting last January.

All of the reports are submitted at this time for approval after having been considered carefully by the respective Divisions and given as wide publicity as possible by publication in the S.A.E. JOURNAL from month to month. The reports as now presented are believed to be in acceptable form, and any proposals should be only in the nature of important and carefully considered constructive changes.

Under the Standards Committee procedure, these reports may be approved as presented, amended within limitations or referred back to the respective Divisions for sufficient reason. The action taken on them by the Standards Committee will be passed upon by the Council and the general business session of the Society and those approved will be published in the S.A.E. HANDBOOK.

Rejection or major changes in any of the reports will require that they be sent back to the Divisions which prepared them and that they cannot be passed upon again before the Annual Meeting of the Society next January. In voting on the reports at the Standards Committee Meeting, the Regulations require that only members of the Standards Committee do so.

Aircraft Division

PERSONNEL

J. F. Hardecker, *Chairman*	Naval Aircraft Factory
Mac Short, *Vice-Chairman*	Stearman Aircraft Co.
Don M. Alexander	Alexander Aircraft Co.
L. Morton Bach	Bach Aircraft Co., Inc.
Harold A. Backus	Berliner-Joyce Aircraft Corp.
Lieut. R. S. Barnaby, U.S.N.	Bureau of Aeronautics, Navy Department
John R. Cautley	Bendix Brake Co.
G. G. Emerson	Wright Aeronautical Corp.
Lieut. C. B. Harper, U.S.N.	Bureau of Aeronautics, Navy Department
H. A. Hicks	Stout Metal Airplane Co.
Major C. W. Howard, U.S.A.	Air Corps
I. M. Laddon	Consolidated Aircraft Corp.
K. M. Lane	Department of Commerce
B. J. Lemon	United States Rubber Co.
Leslie MacDill	Air Corps
C. J. McCarthy	Chance Vought Corp.
J. W. Musser	Aero Supply Mfg. Co., Inc.
C. T. Porter	Keystone Aircraft Corp.
L. D. Seymour	National Air Transport, Inc.
Gerard Vultee	Lockheed Aircraft Co.
Edward Wallace	Great Lakes Aircraft Corp.
E. E. Wilson	Hamilton Standard Propeller Corp.
T. P. Wright	Curtiss Aeroplane & Motor Co., Inc.

Ball Hexagon Bolts and Nuts

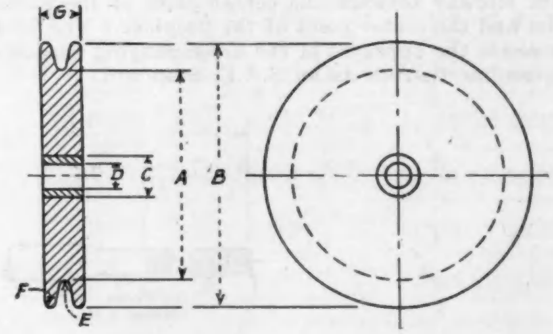
(*Proposed Cancellation of S.A.E. Standard*)

Owing to the fact that the ball hexagon bolts and nuts as covered in the present S.A.E. Standard on pp. 56 and 57 of the 1930 edition of the S.A.E. HANDBOOK are obsolete and no longer in use, the Division recommends the cancellation of these specifications.

Aircraft Pulleys, Plain Bearing

(*Proposed Addition to S.A.E. Standard*)

The Division recommends the inclusion of a 5-in. size of plain pulley corresponding to the one of similar size in the series of anti-friction bearings to have dimensions as shown.

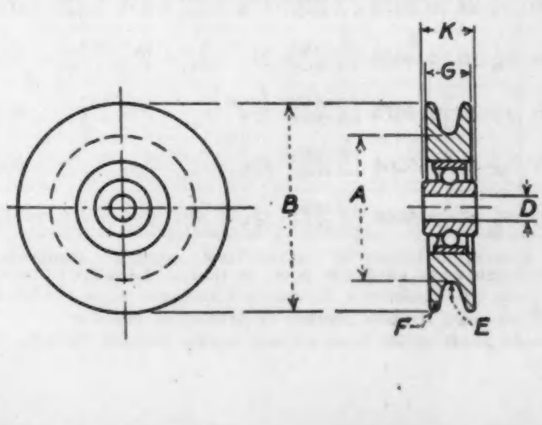


No.	Cable Size	*A*	*B*	*C*	*D* +0.002	*E* Radius	*F* Radius	*G* ±0.015
5	1/8–5/32–3/16	4 3/8	5	1/2	0.378	7/64	3/64	7/16

Pulleys, Non-Metallic, Anti-Friction

(*Proposed S.A.E. Standard*)

A series of anti-friction bearing pulleys similar to the present series of plain-bearing non-metallic pulleys is recommended for approval as an S.A.E. Standard in accordance with the dimensions as shown in the table.



PULLEYS, NON-METALLIC, ANTI-FRICTION

No.	Cable Size	A	B	D	E Radius	F Radius	G ± 0.015	K $^{+0.005}_{-0.000}$
2	1/16–3/32	2	2½	0.191 0.194	3/64	1/32	17/64	0.297
3	1/8–5/32–3/16	1⅜	2	0.251 0.254	7/64	3/64	7/16	0.484
4	1/8–5/32–3/16	2⅞	3½	0.251 0.254	7/64	3/64	7/16	0.484
5	1/8–5/32–3/16	4⅜	5	0.251 0.254	7/64	3/64	7/16	0.484

Note.—Bearing in pulley to withstand side bearing test of 200 lb.

Streamline Tie-Rods

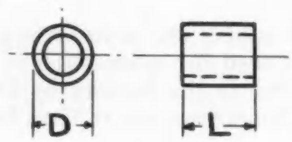
(*Proposed S.A.E. Standard*)

To provide uniformity in dimensions, strength and streamline it has been considered desirable to develop a standard for streamline tie-rods. A variable bend-test for these rods was originally proposed for inclusion but was held in abeyance pending the action of the forthcoming AN Conference on the same item. The dimension K is definite as locating a point midway between the center point of the radius at the slot and the center point of the peephole. The Division recommends the approval of the accompanying specification on streamline tie-rods as an S.A.E. Standard.

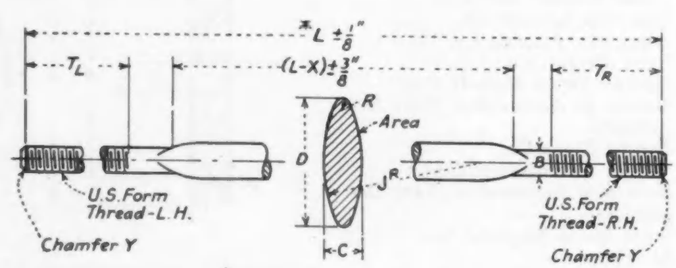
Plain Pulley Spacers

(*Proposed S.A.E. Standard*)

The Division also submits for approval the accompanying spacer dimensions as an S.A.E. Standard.



L	D	Tubing from Which Spacer Is Made	
		Outside Diameter, In.	Size B. w. g.
$0.297^{+0.000}_{-0.005}$	$0.250^{+0.000}_{-0.002}$	¼	22 (0.028)
$0.484^{+0.000}_{-0.005}$	$0.373^{+0.000}_{-0.005}$	⅜	17 (0.058 in.)



Strength, Lb.	Threads[1]	Pitch Diameter	T_L	T_R	X	B	C	D	J	R	Area	Y	K*
1,000	6–40	$0.1218^{+0.0000}_{-0.0017}$	$1\frac{1}{4}{}^{+3/64}_{-1/32}$	$1\frac{3}{4}{}^{+3/64}_{-1/32}$	4	$0.138^{+0.0000}_{-0.0048}$	$0.048^{+0.0024}_{-0.0000}$	0.192	0.288	0.011	$0.0071^{+.0007}_{-.0000}$	45°×1/32	1¼
2,100	10–32	$0.1697^{+0.0000}_{-0.0019}$	$1\frac{3}{8}{}^{+1/16}_{-1/32}$	$1\frac{7}{8}{}^{+1/16}_{-1/32}$	4¼	$0.190^{+0.0000}_{-0.0054}$	$0.064^{+0.0032}_{-0.0000}$	0.256	0.384	0.014	$0.0125^{+.0013}_{-.0000}$	45°×1/32	1½
3,400	¼–28	$0.2268^{+0.0000}_{-0.0022}$	$1\frac{5}{8}{}^{+5/64}_{-1/32}$	$2\frac{1}{8}{}^{+5/64}_{-1/32}$	4¾	$0.250^{+0.0000}_{-0.0062}$	$0.087^{+0.0044}_{-0.0000}$	0.348	0.522	0.019	$0.0234^{+.0023}_{-.0000}$	45°×1/32	1⅞
6,100	5/16–24	$0.2854^{+0.0000}_{-0.0024}$	$1\frac{3}{4}{}^{+5/64}_{-3/64}$	$2\frac{1}{4}{}^{+5/64}_{-3/64}$	5	$0\ 3125^{+0.0000}_{-0.0066}$	$0.110^{+0.0055}_{-0.0000}$	0.440	0.660	0.024	$0.0376^{+.0038}_{-.0000}$	45°×3/64	2⅛
8,000	⅜–24	$0.3479^{+0.0000}_{-0.0024}$	$1\frac{7}{8}{}^{+5/64}_{-3/64}$	$2\frac{3}{8}{}^{+5/64}_{-3/64}$	5¼	$0.375^{+0.0000}_{-0.0066}$	$0.135^{+0.0068}_{-0.0000}$	0.540	0.810	0.030	$0.0563^{+.0056}_{-.0000}$	45°×3/64	2¼
11,500	7/16–20	$0.4050^{+0.0000}_{-0.0026}$	$2\frac{1}{8}{}^{+3/32}_{-3/64}$	$2\frac{5}{8}{}^{+3/32}_{-3/64}$	5¾	$0.4375^{+0.0000}_{-0.0072}$	$0.159^{+0.0080}_{-0.0000}$	0.636	0.954	0.035	$0.0781^{+.0080}_{-.0000}$	45°×3/64	2½
15,500	½–20	$0.4675^{+0.0000}_{-0.0026}$	$2\frac{3}{8}{}^{+3/32}_{-3/64}$	$2\frac{7}{8}{}^{+3/32}_{-3/64}$	6¼	$0.500^{+0.0000}_{-0.0072}$	$0.183^{+0.0092}_{-0.0000}$	0.732	1.098	0.040	$0.1026^{+.0103}_{-.0000}$	45°×3/64	2⅞
20,200	9/16–18	$0.5264^{+0.0000}_{-0.0030}$	$2\frac{5}{8}{}^{+7/64}_{-3/64}$	$3\frac{1}{8}{}^{+7/64}_{-3/64}$	6¾	$0.5625^{+0.0000}_{-0.0082}$	$0.209^{+0.0100}_{-0.0000}$	0.836	1.254	0.045	$0.1354^{+.0135}_{-.0000}$	45°×3/64	3¼
24,700	⅝–18	$0.5889^{+0.0000}_{-0.0030}$	$2\frac{7}{8}{}^{+7/64}_{-1/16}$	$3\frac{3}{8}{}^{+7/64}_{-1/16}$	7¼	$0.625^{+0.0000}_{-0.0082}$	$0.231^{+0.0116}_{-0\ 0000}$	0.924	1.386	0.050	$0.1655^{+.0165}_{-.0000}$	45°×3/64	3½

*To determine length of tie-rod itself, subtract dimension K from length between clevis-pin centers of assembly and specify next longer length when excess is 1/16 in. or more. Lengths to be specified in ¼-in. increments only.

[1] Threads are American Standard (National Fine Pitch) with Class 3 tolerances.

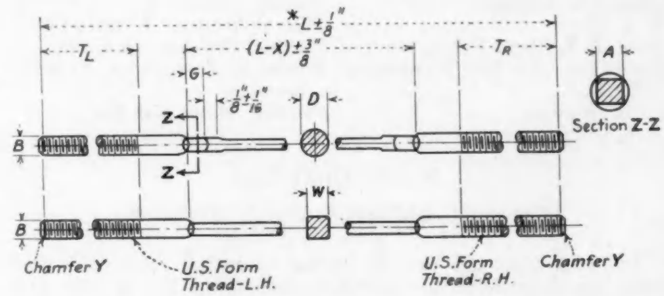
Finished sizes include plating or protective coating.

Tie-rods shall make nine 90-deg. bends without failure, the bends to be back and forth through a total angle of 180 deg.

Internal Tie-Rods

(Proposed S.A.E. Standard)

Similarly the Division recommends the approval of the accompanying specifications on internal tie-rods as an S.A.E. Standard.



Strength, Lb.	Threads[1]	Pitch Diameter	T_L	T_R	X	A	B	D	W	G	Y	K*
1,000	6-40	$0.1218\,{}^{+0.0000}_{-0.0017}$	$1\frac{1}{4}\,{}^{+3/64}_{-1/32}$	$1\frac{3}{4}\,{}^{+3/64}_{-1/32}$	4	0.115	$0.138\,{}^{+0.0000}_{-0.0048}$	$0.101\,{}^{+0.006}_{-0.000}$	$0.089\,{}^{+0.005}_{-0.000}$	3/8	45° × 1/32	1¼
2,100	10-32	$0.1697\,{}^{+0.0000}_{-0.0019}$	$1\frac{3}{8}\,{}^{+1/16}_{-1/32}$	$1\frac{7}{8}\,{}^{+1/16}_{-1/32}$	4¼	0.157	$0.190\,{}^{+0.0000}_{-0.0054}$	$0.134\,{}^{+0.006}_{-0.000}$	$0.118\,{}^{+0.005}_{-0.000}$	3/8	45° × 1/32	1½
3,400	¼-28	$0.2268\,{}^{+0.0000}_{-0.0022}$	$1\frac{5}{8}\,{}^{+5/64}_{-1/32}$	$2\frac{1}{8}\,{}^{+5/64}_{-1/32}$	4¾	0.202	$0.250\,{}^{+0.0000}_{-0.0062}$	$0.180\,{}^{+0.007}_{-0.000}$	$0.155\,{}^{+0.005}_{-0.000}$	½	45° × 1/32	1⅞
6,100	5/16-24	$0.2854\,{}^{+0.0000}_{-0.0024}$	$1\frac{3}{4}\,{}^{+5/64}_{-3/64}$	$2\frac{1}{4}\,{}^{+5/64}_{-3/64}$	5	0.250	$0.3125\,{}^{+0.0000}_{-0.0066}$	$0.223\,{}^{+0.008}_{-0.000}$	$0.201\,{}^{+0.005}_{-0.000}$	½	45° × 3/64	2⅛
8,000	⅜-24	$0.3479\,{}^{+0.0000}_{-0.0024}$	$1\frac{7}{8}\,{}^{+5/64}_{-3/64}$	$2\frac{3}{8}\,{}^{+5/64}_{-3/64}$	5¼	0.300	$0.375\,{}^{+0.0000}_{-0.0066}$	$0.274\,{}^{+0.010}_{-0.000}$	$0.236\,{}^{+0.005}_{-0.000}$	½	45° × 3/64	2¼
11,500	7/16-20	$0.4050\,{}^{+0.0000}_{-0.0026}$	$2\frac{1}{8}\,{}^{+3/32}_{-3/64}$	$2\frac{5}{8}\,{}^{+3/32}_{-3/64}$	5¾	0.360	$0.4375\,{}^{+0.0000}_{-0.0072}$	$0.326\,{}^{+0.012}_{-0.000}$	$0.281\,{}^{+0.005}_{-0.000}$	½	45° × 3/64	2½
15,500	½-20	$0.4675\,{}^{+0.0000}_{-0.0026}$	$2\frac{3}{8}\,{}^{+3/32}_{-3/64}$	$2\frac{7}{8}\,{}^{+3/32}_{-3/64}$	6¼	0.420	$0.500\,{}^{+0.0000}_{-0.0072}$	$0.377\,{}^{+0.014}_{-0.000}$	$0.327\,{}^{+0.005}_{-0.000}$	½	45° × 3/64	2⅞

*To determine length of tie-rod itself, subtract dimension *K* from length between clevis-pin centers of assembly and specify next longer length when excess is 1/16 in. or more. Lengths are to be specified in ¼-in. increments only.

[1] Threads are American Standard (National Fine Pitch) with Class 3 tolerances

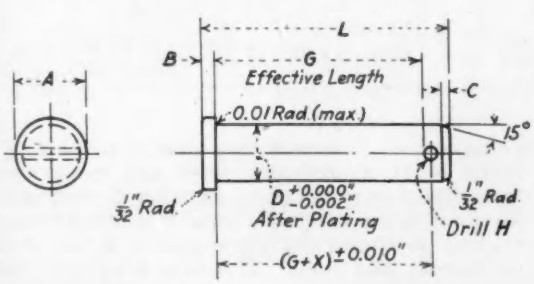
Finished sizes include plating or protective coating.

Tie-rods shall make seven 90-deg. bends without failure, the bends to be back and forth through a total angle of 180 deg.

Flat-Head Pins

(Proposed S.A.E. Standard)

To provide a series of flat-head clevis-pins for use in various aircraft standard fittings the accompanying specification has been developed by the Division with the recommendation that it be approved by the Standards Committee as an S.A.E. Standard.



Nominal Pin Size	A	B	C	D	Drill H	X	Minimum Single Shear Strength —, Lb.
⅛	¼	3/64	3/64	0.124	# 50 (0.070)	0.035	940
3/16	5/16	3/64	3/64	0.186	# 48 (0.076)	0.038	2,130
¼	⅜	1/16	1/16	0.248	# 48 (0.076)	0.038	3,770
5/16	7/16	1/16	5/64	0.311	# 36 (0.1065)	0.053	6,000
⅜	½	1/16	5/64	0.373	# 36 (0.1065)	0.053	8,650
7/16	9/16	3/32	5/64	0.436	# 36 (0.1065)	0.053	11,840
½	⅝	3/32	5/64	0.497	# 36 (0.1065)	0.053	15,400
9/16	11/16	3/32	5/64	0.560	# 28 (0.141)	0.070	19,550
⅝	¾	3/32	5/64	0.622	# 28 (0.141)	0.070	24,150

It is recommended that pins of ⅜ to 7/16 in. in diameter be carried up to 3 in. in length only and that pins ½ to ⅝ in. in diameter be carried up to 4 in. in length; as there is no known demand for greater lengths.

On pins longer than 2 in. the tolerance should be changed from +0.000, —0.002 to +0.000, —0.003.

Pins to be ordered in increments of 1/16 in. only.

Aircraft-Engine Division

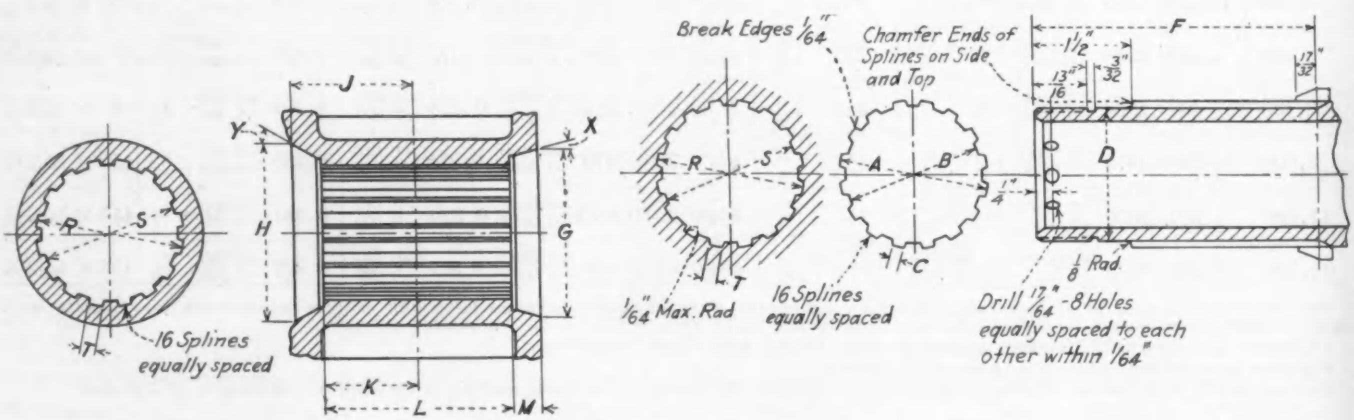
PERSONNEL

Arthur Nutt, *Chairman*	Curtiss Aeroplane & Motor Co., Inc.
L. M. Griffith, *Vice-Chairman*	Emsco Aero Engine Co.
Roland Chilton	Wright Aeronautical Corp.
E. D. Herrick	Lycoming Mfg. Co.
Robert Insley	Continental Aircraft Engine Co.
Ludwig A. Majneri	Warner Aircraft Co.
G. J. Mead	Pratt & Whitney Aircraft Co.
Lieut. E. R. Page, U.S.A.	Materiel Division, Air Corps
Lieut-Com. J. M. Shoemaker, U.S.N.	Bureau of Aeronautics, Navy Department
L. M. Woolson	Packard Motor Car Co.

No. 50 Shaft End

(Proposed Addition to S.A.E. Standard)

In order that the present series of spline shaft-ends and hubs be completed to provide dimensions for shafts and hubs for the larger sizes of engine, it is proposed to adopt new shaft-end and hub dimensions to be known as S.A.E. No. 50, which are similar to those used at present on the shaft end for geared Hornet engines. The accompanying illustrations and table show the proposed dimensions for this new size which it is recommended be approved as an addition to the present S.A.E. Standard on spline shaft-ends and hubs.



PROPOSED NO.-50 HUB DIMENSIONS

Front End *H*	Rear End *G*	*J*	*L*	*M*	*R*	*S*	*T*	For Rear Cone *X*	For Front Cone *Y*
4.562 $^{+0.005}_{-0.000}$	4.625 $^{+0.005}_{-0.000}$	2 13/16	4 15/16	3/4	3.812 $^{+0.005}_{-0.002}$	3.562 $^{+0.005}_{-0.002}$	0.377 ±0.001	15°	30°

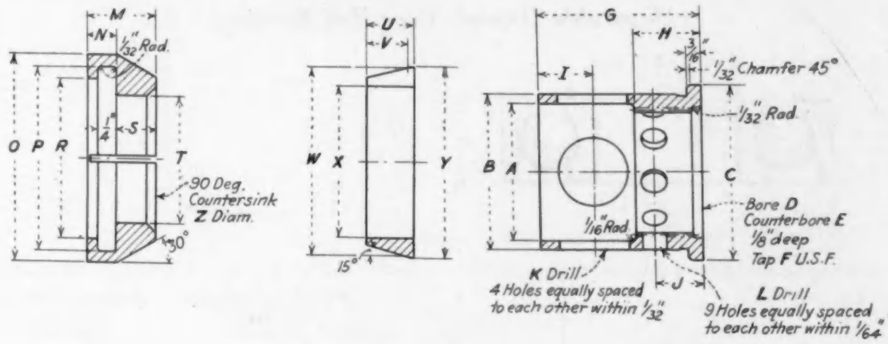
PROPOSED NO.-50 SHAFT-END DIMENSIONS

A	Shaft-End *B*	*C*	Thread	*F*	*R*	*S*	*T*
3.804 $^{+0.000}_{-0.002}$	3.554 $^{+0.000}_{-0.044}$	3.750 ±0.0008	3 7/16—12 U.S.F. P.D. 3.381 $^{+0.000}_{-0.002}$	6 31/32	3.812 $^{+0.005}_{-0.002}$	3.562 $^{+0.005}_{-0.002}$	0.377 ±0.001

Propeller-Hub Cones and Nuts

(Proposed S.A.E. Standard)

Although for some time specifications have existed providing for standard-spline shaft-ends and hubs thereby making these parts interchangeable there has been no interchangeability of cones and nuts from one size of standard shaft-end to another. It has been felt for some time by various engine manufacturers that standardization of these parts would be of considerable value and the matter was submitted to the Aircraft-Engine Division for consideration. As a result the accompanying series of propeller-hub cones and nuts was developed for the complete series of S.A.E. spline shaft-ends and hubs. It is recommended that the Standards Committee approve the accompanying specifications as an S.A.E. Standard.



PROPOSED PROPELLER CONE AND NUT DIMENSIONS

S.A.E. No.	A	B	C	D	E	F	G	H	I	J	K	L
10	1 13/16	2 1/16	2 5/16	1.601 ±0.002	1 45/64	1 11/16—12	2 1/8	7/8	23/32	5/8	57/64	5/16
20	2 3/16	2 7/16	2 11/16	1.976 ±0.002	2 5/64	2 1/16—12	2 1/8	7/8	23/32	5/8	57/64	5/16
30	2 3/8	2 5/8	2 15/16	2.226 ±0.002	2 21/64	2 5/16—12	2 1/8	7/8	23/32	5/8	57/64	5/16
40	2 15/16	3 3/16	3 7/16	2.726 ±0.002	2 7/8	2 13/16—12	2 1/8	7/8	23/32	5/8	57/64	5/16
50	3 15/32	3 3/4	4 3/16	3.351 ±0.002	3 9/16	3 7/16—12	2 7/8	1 9/32	27/32	1 1/16	1 1/32	3/8

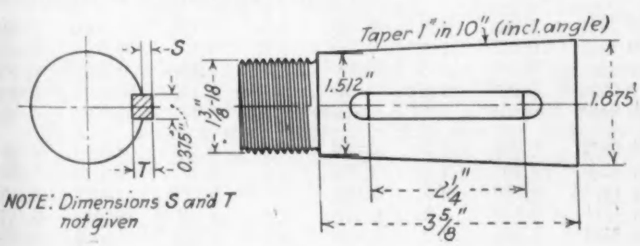
S.A.E. No.	M	N	O	P	R	S	T	U	V	W	X	Y	Z
10	29/32	13/32	2 3/4	2 7/16	2 1/8	17/32	1.6875 ±0.0005	5/8	17/32	2.500	2.001 ±0.0005	2 17/32	1 15/16
20	29/32	13/32	3 1/8	2 13/16	2 1/2	17/32	2.0625 ±0.0005	31/32	17/32	2.875	2.3760 ±0.0005	2 15/16	2 5/16
30	29/32	25/64	3.250 ±0.005	3	2 11/16	1/2	2.3120 ±0.0005	25/32	17/32	3.187	2.626 ±0.0005	3 1/8	2 9/16
40	29/32	13/32	3 7/8	3 9/16	3 1/4	17/32	2.8125 ±0.0005	23/32	17/32	3.625	3.126 ±0.0005	3 11/16	3 1/16
50	1	7/16	4 9/16	4 1/4	3 15/16	9/16	3.5000 ±0.0005	25/32	5/8	4.625	3.8050 ±0.0005	4 11/16	3 9/16

Tapered Shaft Ends

(Proposed Addition to S.A.E. Standard)

During the latter part of last year a small tapered shaft end designated as proposed S.A.E. No. 0 was submitted to the Division for consideration as to its adoption as an addition to the present S.A.E. Standard on spline shaft-ends.

It was stated at the time that this was drawn up, that there are a number of small engines using special tapers which could and should logically use this proposed No. 0. While the number of these engines is not large, it is considered advisable to add this shaft end to the standard in view of the development of many small engines now being designed for use on gliders and small one-place airplanes. It is likely that a still smaller shaft to be known as No. 00 will be developed to provide a suitable shaft for engines of 20 hp. and thereabouts, many of which are now in development. These engines are largely of the two-cylinder opposed type and it will probably be advisable to develop the shaft end before several sizes come into use. Pending this development, however, it is recommended that the Standards Committee approve the addition of the No. 0 size to the present specifications.



Ball and Roller Bearings Division

PERSONNEL

G. R. Bott, *Chairman*	Norma-Hoffmann Bearings Co.
E. R. Carter, *Vice-Chairman*	Fafnir Bearing Co.
H. E. Brunner	S. K. F. Industries, Inc.
F. H. Buhlmann	Rollway Bearing Co., Inc.
D. F. Chambers	Bearings Co. of America
L. A. Cummings	Marlin-Rockwell Corp.
T. C. Delaval-Crow	New Departure Mfg. Co.
H. R. Gibbons	Hyatt Bearings Division, General Motors Corp.
K. L. Herrmann	Bantam Ball Bearing Co.
H. N. Parsons	Strom Bearings Co.
Ernest Wooler	Timken Roller Bearing Co.

Annular Ball Bearings

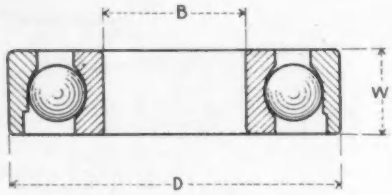
(Proposed Revisions of S.A.E. Standards)

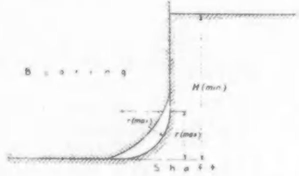
In 1918 the Sectional Committee on Standardization of Ball Bearings was organized by the Society and the American Society of Mechanical Engineers as joint sponsors under the procedure of the then American Engineering Standards Committee to cooperate with the national standardizing bodies in several countries abroad toward arriving insofar as possible at international standardization of anti-friction bearings, the project at that time relating to the radial and thrust types. International communications were maintained and resulted in a revised standard for single-row annular-radial ball-bearings that was approved for adoption as American Standard at the Annual Meeting last January.

The Ball and Roller Bearings Division has since then studied the present standards for the Separable (Open) Type and the Angular Contact Type of Bearings published beginning on p. 288 of the 1930 edition of the S.A.E. HANDBOOK in order to bring them into line with the new tables for the Annular Single-Row Type. The revisions proposed herewith do not affect the basic bores, outside diameters and widths of these bearings for their ranges of sizes but bring the decimal equivalents of metric dimensions, the tolerances, corner radii, radius and eccentricity-tolerance specifications into conformity with the Annular Single-Row Type. The minimum shoulder-heights have been added in these tables.

The accompanying specifications at the time of printing this issue of the S.A.E. JOURNAL were subject to final letter-ballot approval by the Division inasmuch as the meeting of the Division at which they were acted on was not held until April 25.

Separable (Open) Type Ball Bearings





Bearing No.	B		D			W Over-All Width[1]		r=Shift and Housing Fillet Radius, Maximum		H=Height of Shoulder on Shaft, Minimum		Eccentricity Tolerance, Maximum, In.	
	Mm.	In. +0.0000 −0.0004	Mm.	In.	Tolerance, In. −0.0000	Mm.	In. +0.002 −0.002	Mm.	In.	Mm.	In.	Inner Ring	Outer Ring
5	5	0.1069	16	0.6299	+0.0004	5	0.1969	0.2	0.008	0.75	0.029	0.0004	0.0008
6	6	0.2362	21	0.8268	0.0004	7	0.2756	0.3	0.012	1.0	0.039	0.0004	0.0008
7	7	0.2756	22	0.8661	0.0004	7	0.2756	0.3	0.012	1.0	0.039	0.0004	0.0008
8	8	0.3150	24	0.9449	0.0004	7	0.2756	0.3	0.012	1.0	0.039	0.0004	0.0008
9	9	0.3543	28	1.1024	0.0004	8	0.3150	0.3	0.012	1.0	0.039	0.0004	0.0008
10	10	0.3937	28	1.1024	0.0004	8	0.3150	0.3	0.012	2.5	0.098	0.0004	0.0008
11	11	0.4331	32	1.2598	0.0005	7	0.2756	0.4	0.016	2.5	0.098	0.0005	0.0010
12	12	0.4724	32	1.2598	0.0005	7	0.2756	0.4	0.016	2.5	0.098	0.0005	0.0010
13	13	0.5118	30	1.1811	0.0005	7	0.2756	0.3	0.012	2.5	0.098	0.0005	0.0010
14	14	0.5512	35	1.3780	0.0005	8	0.3150	0.5	0.020	2.5	0.098	0.0005	0.0010
15	15	0.5906	35	1.3780	0.0005	8	0.3150	0.5	0.020	2.5	0.098	0.0005	0.0010
16	16	0.6299	38	1.4961	0.0005	10	0.3937	1.0	0.039	3.0	0.118	0.0006	0.0012
17	17	0.6693	44	1.7323	0.0005	11	0.4331	1.0	0.039	3.0	0.118	0.0006	0.0012
18	18	0.7087	40	1.5748	0.0005	9	0.3543	1.0	0.039	3.0	0.118	0.0006	0.0012
19	19	0.7480	40	1.5748	0.0005	9	0.3543	1.0	0.039	3.0	0.118	0.0006	0.0012

[1] The nominal widths of the individual rings shall be the same as the over-all widths given above, but shall have tolerances of plus or minus 0.001 in.

Angular Contact Ball Bearings

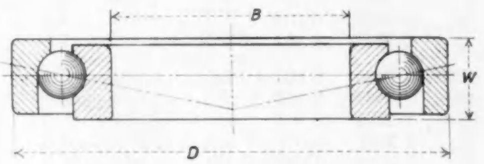
TABLE 1—LIGHT SERIES

Bearing No.	Bore B			Outside Diameter D			Over-All Width W[1]			r Shaft and Housing Fillet Radii,[2] Maximum		H Height of Shoulder on Shaft, Minimum		Eccentricity Tolerances, Maximum, In.	
	Nominal Diameter		Tolerance, In. +0.0000	Nominal Diameter		Tolerance, In. +0.0000	Nominal Width		Tolerance, In., Plus or Minus						
	Mm.	In.		Mm.	In.		Mm.	In.		Mm.	In.	Mm.	In.	Inner Ring	Outer Ring
7200	10	0.3937	0.0004	30	1.1811	0.0004	9	0.3543	0.003	0.6	0.024	2.5	0.098	0.0006	0.0012
7201	12	0.4724	0.0004	32	1.2598	0.0005	10	0.3937	0.003	0.6	0.024	2.5	0.098	0.0006	0.0012
7202	15	0.5906	0.0004	35	1.3780	0.0005	11	0.4331	0.003	0.6	0.024	2.5	0.098	0.0006	0.0012
7203	17	0.6693	0.0004	40	1.5748	0.0005	12	0.4724	0.003	1.0	0.039	3.0	0.118	0.0006	0.0012
7204	20	0.7874	0.0004	47	1.8504	0.0005	14	0.5512	0.003	1.0	0.039	3.0	0.118	0.0006	0.0012
7205	25	0.9843	0.0004	52	2.0472	0.0006	15	0.5906	0.003	1.0	0.039	3.0	0.118	0.0008	0.0012
7206	30	1.1811	0.0004	62	2.4409	0.0006	16	0.6299	0.003	1.0	0.039	3.0	0.118	0.0008	0.0012
7207	35	1.3780	0.0005	72	2.8346	0.0006	17	0.6693	0.003	1.0	0.039	3.5	0.138	0.0008	0.0012
7208	40	1.5748	0.0005	80	3.1496	0.0006	18	0.7087	0.003	1.0	0.039	3.5	0.138	0.0008	0.0012
7209	45	1.7717	0.0005	85	3.3465	0.0008	19	0.7480	0.003	1.0	0.039	3.5	0.138	0.0010	0.0016
7210	50	1.9685	0.0005	90	3.5433	0.0008	20	0.7874	0.003	1.0	0.039	3.5	0.138	0.0010	0.0016
7211	55	2.1654	0.0006	100	3.9370	0.0008	21	0.8268	0.003	1.5	0.059	4.5	0.177	0.0010	0.0016
7212	60	2.3622	0.0006	110	4.3307	0.0008	22	0.8661	0.003	1.5	0.059	4.5	0.177	0.0010	0.0016
7213	65	2.5591	0.0006	120	4.7244	0.0008	23	0.9055	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
7214	70	2.7559	0.0006	125	4.9213	0.0010	24	0.9449	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
7215	75	2.9528	0.0006	130	5.1181	0.0010	25	0.9843	0.005	1.5	0.059	4.5	0.177	0.0010	0.0016
7216	80	3.1496	0.0006	140	5.5118	0.0010	26	1.0236	0.005	2.0	0.079	5.0	0.197	0.0012	0.0018
7217	85	3.3465	0.0008	150	5.9055	0.0010	28	1.1024	0.010	2.0	0.079	5.0	0.197	0.0012	0.0018
7218	90	3.5433	0.0008	160	6.2992	0.0010	30	1.1811	0.010	2.0	0.079	5.0	0.197	0.0012	0.0018
7219	95	3.7402	0.0008	170	6.6929	0.0010	32	1.2598	0.010	2.0	0.079	6.0	0.236	0.0012	0.0018
7220	100	3.9370	0.0008	180	7.0866	0.0010	34	1.3386	0.010	2.0	0.079	6.0	0.236	0.0012	0.0018
7221	105	4.1339	0.0008	190	7.4803	0.0012	36	1.4173	0.010	2.0	0.079	6.0	0.236	0.0012	0.0018
7222	110	4.3307	0.0008	200	7.8740	0.0012	38	1.4961	0.010	2.0	0.079	6.0	0.236	0.0012	0.0018

[1] Width tolerances of individual rings shall be the same as the width tolerances of corresponding sizes of Annular Ball Bearings of the Light, Medium and Heavy Series.

[2] The corner radius or chamfer on bearings must clear the maximum fillet radius given in the table and provide for sufficient bearing area against the minimum shoulder on the shafts.

Conversion from metric to decimal inch dimensions is according to the formula 1 mm. = 0.0393700 in.



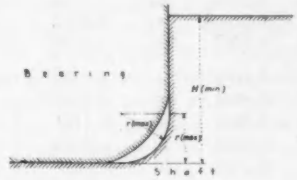


TABLE 2—MEDIUM SERIES

Bearing No.	Bore B			Outside Diameter D			Over-All Width W[1]			r Shaft and Housing Fillet Radii,[2] Maximum		H Height of Shoulder on Shaft, Minimum		Eccentricity Tolerances, Maximum, In.	
	Nominal Diameter		Tolerance, In. +0.0000	Nominal Diameter		Tolerance, In. +0.0000	Nominal Width		Tolerance, In., Plus or Minus					Inner Ring	Outer Ring
	Mm.	In.		Mm.	In.		Mm.	In.		Mm.	In.	Mm.	In.		
7300	10	0.3937	−0.0004	35	1.3780	−0.0005	11	0.4331	0.003	0.6	0.024	2.5	0.098	0.0006	0.0012
7301	12	0.4724	0.0004	37	1.4567	0.0005	12	0.4724	0.003	1.0	0.039	3.0	0.118	0.0006	0.0012
7302	15	0.5906	0.0004	42	1.6535	0.0005	13	0.5118	0.003	1.0	0.039	3.0	0.118	0.0006	0.0012
7303	17	0.6693	0.0004	47	1.8504	0.0005	14	0.5512	0.003	1.0	0.039	3.0	0.118	0.0006	0.0012
7304	20	0.7874	0.0004	52	2.0472	0.0006	15	0.5906	0.003	1.0	0.039	3.5	0.138	0.0006	0.0012
7305	25	0.9843	0.0004	62	2.4409	0.0006	17	0.6693	0.003	1.0	0.039	3.5	0.138	0.0008	0.0012
7306	30	1.1811	0.0004	72	2.8346	0.0006	19	0.7480	0.003	1.0	0.039	3.5	0.138	0.0008	0.0012
7307	35	1.3780	0.0005	80	3.1496	0.0006	21	0.8268	0.003	1.5	0.059	4.5	0.177	0.0008	0.0012
7308	40	1.5748	0.0005	90	3.5433	0.0008	23	0.9055	0.003	1.5	0.059	4.5	0.177	0.0008	0.0012
7309	45	1.7717	0.0005	100	3.9370	0.0008	25	0.9843	0.003	1.5	0.059	4.5	0.177	0.0010	0.0016
7310	50	1.9685	0.0005	110	4.3307	0.0008	27	1.0630	0.003	2.0	0.079	5.0	0.197	0.0010	0.0016
7311	55	2.1654	0.0006	120	4.7244	0.0008	29	1.1417	0.003	2.0	0.079	5.0	0.197	0.0010	0.0016
7312	60	2.3622	0.0006	130	5.1181	0.0010	31	1.2205	0.003	2.0	0.079	6.0	0.236	0.0010	0.0016
7313	65	2.5591	0.0006	140	5.5118	0.0010	33	1.2992	0.005	2.0	0.079	6.0	0.236	0.0010	0.0016
7314	70	2.7559	0.0006	150	5.9055	0.0010	35	1.3780	0.005	2.0	0.079	6.0	0.236	0.0010	0.0016
7315	75	2.9528	0.0006	160	6.2992	0.0010	37	1.4567	0.005	2.0	0.079	6.0	0.236	0.0010	0.0016
7316	80	3.1496	0.0006	170	6.6929	0.0010	39	1.5354	0.005	2.0	0.079	6.0	0.236	0.0012	0.0018
7317	85	3.3465	0.0008	180	7.0866	0.0010	41	1.6142	0.010	2.5	0.098	7.0	0.276	0.0012	0.0018
7318	90	3.5433	0.0008	190	7.4803	0.0012	43	1.6929	0.010	2.5	0.098	7.0	0.276	0.0012	0.0018
7319	95	3.7402	0.0008	200	7.8740	0.0012	45	1.7717	0.010	2.5	0.098	7.0	0.276	0.0012	0.0018
7320	100	3.9370	0.0008	215	8.4646	0.0012	47	1.8504	0.010	2.5	0.098	7.0	0.276	0.0012	0.0018
7321	105	4.1339	0.0008	225	8.8583	0.0012	49	1.9291	0.010	2.5	0.098	7.0	0.276	0.0012	0.0018
7322	110	4.3307	0.0008	240	9.4488	0.0012	50	1.9685	0.010	2.5	0.098	7.0	0.276	0.0012	0.0018

[1]Width tolerances of individual rings shall be the same as the width tolerances of corresponding sizes of Annular Ball Bearings of the Light, Medium and Heavy Series.

[2]The corner radius or chamfer on bearings must clear the maximum fillet radius given in the table and provide for sufficient bearing area against the minimum shoulder on the shafts.

Conversion from metric to decimal inch dimensions is according to the formula 1 mm. = 0.0393700 in.

TABLE 3—HEAVY SERIES

Bearing No.	Bore B			Outside Diameter D			Over-All Width W[1]			r Shaft and Housing Fillet Radii,[2] Maximum		H Height of Shoulder on Shaft, Minimum		Eccentricity Tolerances, Maximum, In.	
	Nominal Diameter		Tolerance, In. +0.0000	Nominal Diameter		Tolerance, In. +0.0000	Nominal Width		Tolerance, In., Plus or Minus					Inner Ring	Outer Ring
	Mm.	In.		Mm.	In.		Mm.	In.		Mm.	In.	Mm.	In.		
7403	17	0.6693	−0.0004	62	2.4409	−0.0006	17	0.6693	0.003	1.0	0.039	4.5	0.177	0.0006	0.0012
7404	20	0.7874	0.0004	72	2.8346	0.0006	19	0.7480	0.003	1.0	0.039	4.5	0.177	0.0006	0.0012
7405	25	0.9843	0.0004	80	3.1496	0.0006	21	0.8268	0.003	1.5	0.059	5.0	0.197	0.0008	0.0012
7406	30	1.1811	0.0004	90	3.5433	0.0008	23	0.9055	0.003	1.5	0.059	5.0	0.197	0.0008	0.0012
7407	35	1.3780	0.0005	100	3.9370	0.0008	25	0.9843	0.003	1.5	0.059	5.0	0.197	0.0008	0.0012
7408	40	1.5748	0.0005	110	4.3307	0.0008	27	1.0630	0.003	2.0	0.079	5.5	0.217	0.0008	0.0012
7409	45	1.7717	0.0005	120	4.7244	0.0008	29	1.1417	0.003	2.0	0.079	5.5	0.217	0.0010	0.0016
7410	50	1.9685	0.0005	130	5.1181	0.0010	31	1.2205	0.003	2.0	0.079	6.5	0.256	0.0010	0.0016
7411	55	2.1654	0.0006	140	5.5118	0.0010	33	1.2992	0.003	2.0	0.079	6.5	0.256	0.0010	0.0016
7412	60	2.3622	0.0006	150	5.9055	0.0010	35	1.3780	0.003	2.0	0.079	6.5	0.256	0.0010	0.0016
7413	65	2.5591	0.0006	160	6.2992	0.0010	37	1.4567	0.005	2.0	0.079	6.5	0.256	0.0010	0.0016
7414	70	2.7559	0.0006	180	7.0866	0.0010	42	1.6535	0.005	2.5	0.098	7.5	0.295	0.0010	0.0016
7415	75	2.9528	0.0006	190	7.4803	0.0012	45	1.7717	0.005	2.5	0.098	7.5	0.295	0.0010	0.0016
7416	80	3.1496	0.0006	200	7.8740	0.0012	48	1.8898	0.005	2.5	0.098	7.5	0.295	0.0012	0.0018
7417	85	3.3465	0.0008	210	8.2677	0.0012	52	2.0472	0.010	3.0	0.118	9.5	0.374	0.0012	0.0018
7418	90	3.5433	0.0008	225	8.8583	0.0012	54	2.1260	0.010	3.0	0.118	9.5	0.374	0.0012	0.0018

[1]Width tolerances of individual rings shall be the same as the width tolerances of corresponding sizes of Annular Ball Bearings of the Light, Medium and Heavy Series.

[2]The corner radius or chamfer on bearings must clear the maximum fillet radius given in the table and provide for sufficient bearing area against the minimum shoulder on the shafts.

Conversion from metric to decimal inch dimensions is according to the formula 1 mm. =0.0393700 in.

Electrical Equipment Division

PERSONNEL

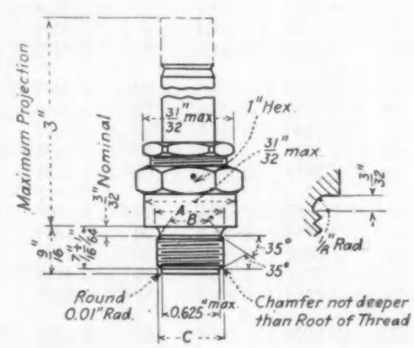
D. M. Pierson, *Chairman*	Chrysler Corp.
D. S. Cole, *Vice-Chairman*	Leece-Neville Co.
Azel Ames	Kerite Insulated Wire & Cable Co.
A. K. Brumbaugh	White Motor Co.
W. S. Haggott	Packard Electric Co.
L. M. Kanters	Waukesha Motor Co.
T. L. Lee	North East Appliance Corp.
A. R. Lewellen	Chevrolet Motor Co.
L. E. Lighton	Electric Storage Battery Co.
L. O. Parker	Delco-Remy Corp.
E. K. Schadt	Cadillac Motor Car Co.
T. E. Wagar	Studebaker Corp.

Metric Spark-Plugs for Automobiles

(Proposed S.A.E. Standard)

The Standards Committee at its last meeting in January referred back to this Division and the Subdivision on Metric Spark-Plugs, the report submitted at that time because of conflicting opinions on some of the dimensions submitted.

Since that time this matter has been thrashed out in a Subdivision meeting and a new specification drawn up. While there was some opposition to this on the part of two of the Subdivision members, the dimensions as submitted were approved for submission to the Electrical Equipment and Gasoline Engine Divisions. The letter ballot taken on this shows an overwhelming approval of the specifications on the part of both Divisions and on the basis of this it is recommended that the Standards Committee approve the accompanying specifications on metric spark-plugs as an S.A.E. Standard.



Thread—18-mm. diameter; pitch 1½ mm.

Form of thread—International standard (same as American Standard except ½ as much truncation at root of thread).

SPARK-PLUG DIMENSIONS

Dimensions	Maximum		Minimum	
	Mm.	In.	Mm.	In.
A	18.50	0.728	18.00	0.709
B	.....	0.633[1]	.....	0.625
C	17.97	0.708	17.85	0.703

SPARK-PLUG THREAD TOLERANCES

Diameter	Maximum		Minimum	
	Mm.	In.	Mm.	In.
Outside (Full)	17.975	0.70768 (0.708)	17.850	0.70275 (0.703)
Pitch (Effective)	17.001	0.66933 (0.669)	16.876	0.66441 (0.664)
Root (Core)	15.864	0.62457 (0.625)	15.739	0.61964 (0.620)

TAPPED-HOLE THREAD TOLERANCES

Diameter	Maximum		Minimum	
	Mm.	In.	Mm.	In.
Outside (Full)	18.312	0.72094 (0.721)	18.187	0.71602 (0.716)
Pitch (Effective)	17.176	0.67622 (0.676)	17.051	0.67129 (0.671)
Root (Core)	16.201	0.63783 (0.638)	16.076	0.63291 (0.633)

Lighting Division

PERSONNEL

C. A. Michel, *Chairman*	Guide Lamp Corp.
R. E. Carlson, *Vice-Chairman*	Westinghouse Lamp Co.
C. C. Bohner	Tung-Sol Lamp Works
A. W. Devine	Massachusetts Registry of Motor Vehicles
H. C. Doane	Buick Motor Co.
G. P. Doll	Thomas J. Corcoran Lamp Co.
R. N. Falge	General Motor Corp.
W. S. Hadaway	Edison Lamp Works of the General Electric Co.
C. L. Holm	Ford Motor Co.
R. W. Johnson	John W. Brown Mfg. Co.
W. M. Johnson	National Lamp Works of the General Electric Co.
A. R. Lewellen	Chevrolet Motor Co.
A. L. Martinek	C. M. Hall Lamp Co.
D. M. Pierson	Chrysler Corp.
W. F. Thoms	Allied Products Corp.
T. E. Wagar	Studebaker Corp.

Automobile Headlighting Specifications

(Proposed Revision of S.A.E. Standard)

Since the present specifications for Laboratory Tests printed on p. 134 of the 1930 edition of the S.A.E. HANDBOOK were adopted by the Society last January, some changes were suggested to the Lighting Division by one of its members. They were also sent to the Motor-Vehicle Lighting Committee of the Illuminating Engineering Society which has been cooperating with the Lighting Division in this work for a number of years. At a meeting of this Committee in Schenectady on April 15 at which Chairman Michel and three other members of the Lighting Division were present, the suggested changes were discussed at considerable length and a number of recommendations approved unanimously. These recommendations were referred to the Lighting Division of the Standards Committee for their approval of the following proposed changes in the specification and are printed in this issue of the S.A.E. JOURNAL subject to final approval by the Division letter-ballot.

Specifications for Laboratory Tests of Optical Characteristics of Electric Head-Lamps for Motor-Vehicles

In the paragraph entitled *Scope of Specification*, printed on p. 134 of the 1930 edition of the S.A.E. HANDBOOK, insert "automobile" before the word "head-lamp" at the end of the paragraph making it read "This specification is intended to cover the optical performance of present conventional types of automobile head-lamp."

Transfer *Scope of Specification* paragraph to precede that entitled *Definitions* and add the following paragraph:

> In view of the fact that the headlighting art is a developing one, these specifications are necessarily of a temporary character and are subject to revision from time to time. It follows, therefore, that while they are applicable to use in connection with regulation by State authorities having administrative powers, they are not suitable for inclusion in State laws where the requisite flexibility of revision is absent.

In the paragraph immediately following the heading Specifications for Laboratory Tests of Head-Lamps on p. 136 of the S.A.E. HANDBOOK, delete "printed at the bottom of the other column" and insert "following" before the word "limitations" at the end of the paragraph.

Delete the reference to and limitations for auxiliary driving-lamps on p. 137 of the S.A.E. HANDBOOK.

Electric Signal Lamps

(Proposed S.A.E. Standard)

The S.A.E. Recommended Practice on Signal Lamps that was published in the 1928 edition of the S.A.E. HANDBOOK was found unsatisfactory and cancelled. A Subdivision of the Lighting Division was appointed to redraft this specification. This Subdivision, working jointly with a Subcommittee of the Motor-Vehicle Lighting Committee of the Illuminating Engineering Society, secured a signal lamp set-up and conducted tests on illumination characteristics of signal lamps for lenses of different colors and varying area of illuminated surface, both with and without an opaque figure on the lens. A report was then submitted at a joint meeting of the Illuminating Engineering Society Committee and the Lighting Division in Detroit last December but several points in the report were not accepted. A revised report was accordingly submitted at the meeting of the Motor-Vehicle Lighting Committee at Schenectady on April 15 that was approved subject to approval by the Lighting Division. The report was accordingly referred to the Lighting Division for approval by letter-ballot and at the time of printing it was subject to the Division's approval by ballot.

Specifications for Laboratory Tests of Optical Characteristics of Electric Signal Lamps for Motor-Vehicles

(Proposed Revised S.A.E. Recommended Practice)

Definition.—Signal lamp, a device used to indicate the intention of the operator of a motor-vehicle to diminish speed, stop, or change direction.

Scope of Specification.—This specification applies only to electric signal lamps whose attracting power depends primarily on brightness. It does not apply to signals embodying mechanical motion, flashing mechanisms, etc.

Samples for Test.—Sample signal lamps submitted for laboratory test shall be representative of the devices as regularly manufactured and marketed. Each sample shall include incandescent lamps and any other accessory equipment peculiar to the device and necessary to operate it in its normal manner. Each sample shall be accompanied by any instructions necessary for determining its normal position when mounted on the vehicle.

Set-Up for Testing.—The laboratory shall be equipped with all facilities necessary to make accurate photometric measurements in accordance with established laboratory practices.

Unless otherwise specified, the incandescent lamps used in signal-lamp tests shall be supplied by the laboratory. They shall be representative of standard incandescent lamps in regular production for automotive service. They shall be selected for accuracy in accordance with specifications approved by the Bureau of Standards. They shall be operated at their rated mean spherical candlepower during the tests. Where special incandescent lamps are specified, such lamps shall be submitted with the devices and the same or similar lamps used in the tests and operated at their rated mean spherical candlepower.

All beam-candlepower measurements shall be made with the center of light at a distance of 4 ft. from the photometer screen. In measuring distances and angles, the incandescent filament shall be taken as the center of light.

Photometric Test Requirements.—A stop-signal indication shall be either red or amber. When measured in a plane normal to the lamp axis, the projected luminous area of the signal, plus any opaque areas entirely within this luminous

area, shall not be less than 3½ sq. in. The projected luminous area alone shall not be less than 2 sq. in.

On a horizontal line through the light source and parallel to the longitudinal axis of the vehicle, the average brightness of the luminous signal area shall not be less than 2 cp. per sq. in. and the intensity shall not be less than 7 cp.

At all angles within 30 deg. to the left and right of the vertical plane through the lamp axis, and in a plane at 90 deg. to this vertical plane which passes through the light source and is 15 deg. above the lamp axis, the intensity shall not be less than 0.5 cp.

Motorcoach and Motor Truck Division

PERSONNEL

Arthur W. Herrington, *Chairman*	Coleman Motors Corp.
A. Gelpke, *Vice-Chairman*	Autocar Co.
W. J. Baumgartner	Relay Motors Corp.
M. C. Horine	International Motor Co.
L. C. Josephs	International Motor Co.
F. W. Kateley	American Car & Foundry Motors Co.
W. F. Klein	General Motors Truck Co.
A. A. Lyman	Public Service Coordinated Transport
J. A. Packard	Studebaker Corp.
L. H. Palmer	Fifth Ave. Coach Co.
A. J. Scaife	White Motor Co.
E. M. Schultheis	Timken Roller Bearing Co.
P. V. C. See	Northern Ohio Power & Light Co.
Ernest M. Sternberg	Sterling Motor Truck Co.

Motorcoach Specifications

(*Proposed Cancellation of S.A.E. Recommended Practice*)

In 1925 the present Motorcoach Specifications, printed on p. 603 of the 1930 edition of the S.A.E. HANDBOOK, were adopted as a guide to State officials in formulating regulations that would be sufficiently uniform in their requirements to place no serious handicap on motorcoach manufacturers or operators or restrict proper development of this class of vehicles.

An informal Committee, appointed by the Motor Truck Committee of the National Automobile Chamber of Commerce in May, 1926, drafted a uniform motorcoach specifications code that included all items of motorcoach construction which would ordinarily be included in State regulations and indicated those which the Committee felt are the only ones that should be governed by State regulations. The final draft of the code was approved by a general conference of manufacturing and operating interests at the City of Washington in June, 1929, and referred to the Society, among others, for supporting action. The Transportation and Maintenance Committee of the Society studied the code and at its meeting in Detroit last January, recommended that the present S.A.E. Motorcoach Specifications be cancelled until revised specifications that will meet modern requirements are adopted by the Society. This recommendation was referred to the Motorcoach and Motor Truck Division of the Standards Committee which recommends by unanimous letter-ballot that the present S.A.E. Recommended Practice for Motorcoach Specifications be cancelled.

Non-Ferrous Metals Division

PERSONNEL

Zay Jeffries, Chairman	Aluminum Company of America
C. W. Simpson, Vice-Chairman	White Motor Co.
R. J. Allen	Rolls Royce of America, Inc.
W. H. Bassett	American Brass Co.
C. H. Calkins	Baush Machine Tool Co.
D. L. Colwell	Stewart Die Casting Corp.
H. R. Corse	Lumen Bearing Co.
W. A. Cowan	National Lead Co.
R. M. Curts	New Jersey Zinc Co.
P. V. Faragher	Aluminum Company of America
W. H. Graves	Packard Motor Car Co.
H. L. Greene	Willys-Overland, Inc.
J. B. Johnson	Air Corps
R. R. Moore	Wright Aeronautical Corp.
H. C. Mougey	General Motors Corp.
Dr. W. B. Price	Scovill Manufacturing Co.
T. H. Wickenden	International Nickel Co.
H. M. Williams	Frigidaire Corp.

Aluminum Alloy Specifications

(*Proposed Revision of and Additions to S.A.E. Standard*)

The Subdivision on Light Metals and Alloys, through its chairman, P. V. Faragher, submitted to the Non-Ferrous Metals Division a revision of several of the present S.A.E. specifications for aluminum alloys and a series of specifications for new alloys not heretofore covered by S.A.E. specifications. No determination has as yet been made of numbers for the new alloys, this being a matter that will, of necessity, be taken up by the Division before publication of the specifications.

The Division recommends approval by the Standards Committee of the revisions and additional aluminum alloy specifications as follows:

SPECIFICATION NO. 36

Composition, in Percentage

Copper	7.0—8.5
Silicon	1.0—1.5
Iron	0.8—1.4
Zinc	not over 0.2
Manganese	not over 0.3
Magnesium	not over trace
Other impurities	not over 0.3
Aluminum	Remainder

General Information—Standard tensile test specimens cast in sand and tested without machining give tensile strengths in the range 19,000 to 24,000 lb. per sq. in. The elongation is usually from 1 to 3 per cent in 2 in., but because of the difficulty of measuring elongations in this range with accuracy it is customary to omit elongation requirements from specifications for this alloy.

This alloy was developed from alloy No. 30 and differs from it only in that iron and silicon are added in carefully controlled amounts instead of being allowed to vary at random in the range permitted by the limit on impurities. In this alloy, iron and silicon are definite alloying constituents and are added to improve the casting characteristics of the old aluminum-copper casting alloy. With this alloy, the tendency for cracks and shrinks is very greatly decreased. It has largely superseded No. 30, particularly for more difficult castings that require pouring temperatures higher than are usually used.

This is a light alloy having a specific gravity of about 2.83 and is commonly used where alloy No. 30 has previously been specified. A shrinkage of 0.156 (5/32) in. per ft. should be allowed in pattern design. This alloy and No. 33 are most commonly used for casting crankcases, oil pans, differential carriers, transmission cases and other such castings.

SPECIFICATION NO. 30

In the note to this alloy, add the following:

For more difficult castings, alloy No. 36, which has similar mechanical properties and superior casting characteristics, is now commonly specified.

SPECIFICATION NO. 31A

Composition, in Percentage

Copper	2.0 — 3.5
Zinc	9.0 —11.5

Iron	1.25— 1.75
Total Other Impurities	not over 1.0
Aluminum	Remainder

Test specimens cast in sand and tested without machining give tensile strengths in the range 25,000 to 36,000 lb. per sq. in. and elongations from 3 to 7 per cent in 2 in.

The specific gravity is about 3.0. A shrinkage of 0.156 (5/32) in. per ft. should be allowed in pattern designs.

This alloy is used where somewhat higher mechanical properties are desired than are obtained with alloys No. 30, 33 and 36. In common with other alloys containing considerable percentages of zinc, this alloy loses its strength more rapidly as the temperature is raised than do some of the other alloys, hence it is not recommended for use at elevated temperatures.

Specification No. 37

Composition, in Percentage

Silicon	12.0—13.0
Iron	not over 0.8
Copper	not over 0.3
Zinc	not over 0.2
Manganese	not over 0.5
Magnesium	not over trace
Total Other Impurities	not over 0.3
Aluminum	Remainder

General Information—Standard test bars, cast to size in sand, show tensile strengths from 24,000 to 31,000 lb. per sq. in. and elongations from 5.0 to 15.0 per cent. These properties are obtained only if the molten alloy is subjected to a special process, known as "modification," immediately before it is poured. The specific gravity is 2.68.

Like alloy No. 35, this alloy is especially resistant to salt-water corrosion. Because of its good foundry characteristics it can be used for complicated castings consisting of both thin and heavy sections. The ratio of yieldpoint to ultimate tensile-strength is somewhat lower than is the case with some of the other aluminum casting-alloys, which fact should be considered in connection with the properties shown in the preceding paragraph.

Specification No. 38 (Heat-Treated Castings)

Composition, in Percentage

Copper	4.0—5.0
Silicon	not over 1.20
Iron	not over 1.20
Zinc	not over 0.25
Total of all constituents except aluminum and copper	not over 2.5

General Information—Castings from this alloy can be heat-treated to produce mechanical properties distinctly higher than those of the ordinary aluminum casting-alloys. The heat-treatment may be varied depending upon the service requirements.

Where maximum toughness and resistance to shock are desired, a solution heat-treatment alone is used, which produces in sand-cast test-specimens tensile strengths from 28,000 to 38,000 lb. per sq. in. and elongations from 6 to 12 per cent in 2 in. On standing at room temperatures, there is an aging effect which is practically complete in a few months. The tensile strength increases by a few thousand pounds per square inch, and the elongation decreases by a few per cent in 2 in. The greatest change occurs in the yieldpoint, which increases to a value usually well in excess of 20,000 lb. per sq. in.

If somewhat greater initial strength, hardness, and yield-point are desired, the solution heat-treatment may be followed by a precipitation heat-treatment in which tensile strengths from 30,000 to 40,000 lb. per sq. in. and elongations from 3 to 8 per cent in 2 in. are developed.

If still greater strength and hardness are desired, the heat-treatment may be varied to produce tensile strengths from 36,000 to 50,000 lb. per sq. in. and elongations from 0 to 5 per cent in 2 in.

Since the introduction of this alloy, its use has developed into very substantial tonnages. It is used in windshield frames, fire-engine, motorcoach-engine and aircraft-engine crankcases, and for a variety of parts in both motor-vehicle and aircraft assemblies, where a high-strength light-weight casting is desired. It is very much more resistant to corrosion than the ordinary alloys containing copper and is extensively used in the manufacture of outboard motors and for castings to be used on board ships.

Specification No. 39

Composition, in Percentage

Copper	3.75—4.25
Nickel	1.8 —2.3
Magnesium	1.2 —1.7
Iron	not over 1.0
Silicon	not over 0.7
Total other impurities	not over 0.2
Aluminum	Remainder

General Information—Standard test-specimens, cast to size in sand, after proper heat-treatment show tensile strengths from 30,000 to 42,000 lb. per sq. in. and elongations from 0 to 2.0 per cent. If the alloy is not heat-treated, the tensile strength is about one-third less than these values. Because of its property of retaining its strength better at elevated temperatures than do many of the other aluminum alloys, it is used for pistons and cylinder-heads of aircraft engines and for other parts where this property is an advantage.

Alloy No. —

Composition, in Percentage

Copper	3.5—4.5
Magnesium	0.2—0.75
Manganese	0.4—1.0
Aluminum	not less than 92.0

General Information—This alloy is the alloy commonly designated "duralumin," "Dural" or "17S." It is used in the wrought condition either rolled, drawn, forged or extruded. In the form of sheet, this alloy is supplied to the following mechanical property requirements:

DURALUMIN OR 17S SHEET, HEAT-TREATED

Thickness, In.	Minimum Tensile Strength, Lb. per Sq. In.	Minimum Yield-Point, Lb. per Sq. In.	Minimum Elongation in 2 In. Per Cent
0.010 to 0.020	55,000	30,000	15
0.021 to 0.128	55,000	30,000	18[a]
0.129 to 0.258	55,000	30,000	15
0.259 to 0.500	55,000	30,000	12
0.501 to 1.75	55,000	30,000	9

[a] Sheets 30 in. or more in width in thicknesses of 0.021 to 0.040 in. may show a minimum elongation of 17 per cent.

In the annealed temper, the tensile strength shall not exceed 35,000 lb. per sq. in.

The following thickness tolerances are commercial for this product:

THICKNESS TOLERANCES FOR DURALUMIN OR 17S SHEET

Thickness, In.	Plus or Minus Tolerance, In. Widths 18 In. and Less	Over 18 In. to 36 In.	Over 36 In.
0.013 to 0.036	0.0015	0.002	0.0025
0.037 to 0.050	0.002	0.0025	0.003
0.051 to 0.070	0.0025	0.003	0.004
0.071 to 0.090	0.003	0.003	0.004
0.092 and heavier	5% T	5% T	....

(T = nominal thickness of sheet.)

In the form of tubing and extruded shapes, the same tensile strength and yield-point are obtained, the elongation varying with the cross-sectional area of the test section. Rod and bar will also show these same properties, except in certain of the larger sizes where manufacturing limitations may result in slightly lower values.

This alloy is extensively used in the construction of motorcoaches, trucks, and aircraft, both heavier and lighter than air. It is available in all the forms in which metals are commonly used: sheet, plate, strip, tubing (including streamline and other special shapes), bar, rod, wire, rivets, bolts, nuts, screws and other screw-machine products, structural shapes and extruded sections specially designed for aircraft construction.

Alloy No. —

Composition Limits, in Percentage

Copper	3.9—5.0
Manganese	0.5—1.1
Silicon	0.5—1.1
Aluminum	not less than 92.0

General Information.—This alloy is known as 25S and, like duralumin, is used in the wrought condition. In the fully heat-treated temper, the tensile strength is from 55,000 to 63,000 lb. per sq. in. The elongation, except for very thin or very heavy sections, is between 16 and 22 per cent in 2 in. If the alloy be given the solution heat-treatment, but not the precipitation heat-treatment, it is in a more workable condition and has the following properties:

Tensile Strength, 45,000 to 53,000 lb. per sq. in.

Elongation, 15 to 22 per cent in 2 in.

Yield-Point, 15,000 to 30,000 lb. per sq. in.

In the annealed temper, the tensile strength varies from 25,000 to 35,000 lb. per sq. in.

This alloy is commonly available in a variety of forms including sheet, plate, bar and structural shapes. It is especially adapted for the manufacture of forgings, because of its excellent hot-working properties. Forgings of this alloy in common use include connecting-rods for automobile engines, propellers for aircraft, automobile hardware and fittings of various types.

Alloy No. —

Composition Limits, in Percentage

Magnesium	0.45—0.9
Silicon	0.6 —1.2
Aluminum	not less than 96.3

General Information.—This alloy is designated in the trade as 51S. It is used in the wrought form and is one of the alloys susceptible to heat-treatment. In the fully heat-treated temper (designated as T), the tensile strength is from 45,000 to 50,000 lb. per sq. in.; the elongation varies from 10 to 18 per cent in 2 in. except in very thin or very heavy sections. The yield-point is from 30,000 to 40,000 lb. per sq. in.; substantially the same as that of duralumin or 25S, although the ultimate strength is about 10,000 lb. lower than that of these alloys. If the alloy is not subjected to the precipitation heat-treatment after it has been heated and quenched, it has a tensile strength in the range 30,000 to 40,000 lb. per sq. in. and an elongation of 20 to 30 per cent in 2 in. In this temper (designated W) the alloy can be formed with considerable ease. In the annealed temper, the tensile strength varies from 14,000 to 19,000 lb. per sq. in. and the elongation from 22 to 32 per cent in 2 in.

This alloy is available in practically all the wrought forms in which aluminum and other metals are produced; these include sheet, plate, tubing, rod, bar, wire, shapes, moldings, both rolled and extruded, forgings, and so forth.

Because of the readiness with which this alloy can be worked hot, it is used for the manufacture of complicated forgings, such as radial aircraft-engine crankcases. It is also used for less highly stressed parts, such as automobile hardware.

Alloy No. —

Composition Limits, in Percentage

Manganese	1.0—1.5
Copper	not over 0.2
Aluminum, minimum	97.0

General Information.—This alloy is used to replace pure aluminum where somewhat higher tensile strength and hardness are desired. It is not quite so easily formed as is commercial aluminum in the corresponding tempers. This alloy shows substantially the same resistance to atmospheric corrosion and to most forms of chemical attack as does aluminum of commercial purity.

As is the case with commercial aluminum, the various tempers are obtained by varying the amount of cold work done on the alloy, after its final annealing. It is regularly manufactured in the form of sheet, plate, tubing, rod, bar and wire, extruded shapes, moldings, and so on.

The alloy in the various standard tempers has the tensile strengths shown in the following table. The elongation values for flat sheet are also shown. In the case of strip sheet in the half-hard temper, the elongation may be slightly lower than the values shown for flat sheet.

Temper	Tensile Strength, Minimum, Lb. per Sq. In.	Elongation in 2 In., Per Cent Thickness, In. 0.013 to 0.019	0.020 to 0.031	0.032 to 0.050	0.051 to 0.113	0.114 to 0.162	0.163 to 0.258	0.259 to 0.375
Soft	14,000	20	20	23	25	25	25	28
One-half Hard	19,500	3	4	5	6	7	8	8
Three-fourths Hard	24,000	1	2	3	4			
Hard	27,000	1	2	3	4			

The thickness tolerances in the following tables are met in commercial manufacture of sheet.

THICKNESS TOLERANCES FOR FLAT SHEET

B. & S. Gage	Thickness, In.	Width 20 In. and Under	Over 20 In. to 36 In. Incl.	Over 36 In. to 60 In. Incl.
2 and 3	0.250[b]—0.220	±0.007	±0.008	±0.009
4 to 9	0.219 —0.115	±0.005	±0.006	±0.007
10 to 13	0.114 —0.073	±0.003	±0.0035	±0.004
14 to 21	0.072 —0.030	±0.0025	±0.0025	±0.003
22 to 24	0.029 —0.019	±0.002	±0.002	±0.003
25 to 27	0.018 —0.014	±0.002	±0.002	±0.002
28 to 36	0.013 —0.005	±0.0015	±0.0015	±0.0015

[b] Material having thickness greater than 0.250 in. is considered plate and is covered by plate tolerances.

THICKNESS TOLERANCES FOR STRIP SHEET

B. & S. Gage	Thickness, In.	Width: ¼ In. to 12 In., Incl.	Width: Over 12 In. to 24 In., Incl.
10 to 18	0.102—0.041	±0.003	±0.003
19 and 20	0.040—0.030	±0.002	±0.0025
21 to 25	0.029—0.017	±0.002	±0.0025
26 to 29	0.016—0.011	±0.0015	....
30 to 36	0.010—0.005	±0.001	....

SPECIFICATION No. 33

Composition, in Percentage

Copper	6.0—8.0
Zinc	not over 2.5
Iron	not over 1.5
Silicon	not over 1.0
Total other impurities	not over 1.0
Aluminum	Remainder

Test specimens cast in sand and tested without machining show tensile strengths in the range 19,000 to 24,000 lb. per sq. in. and elongation from 1.0 to 2.5 per cent in 2 in. It is customary to omit elongation requirements from specifications for this alloy.

General Information.—This is a light alloy, having a specific gravity of 2.83 to 2.86, and has been the most extensively used alloy in the automotive industry. It is similar to Specifications Nos. 30 and 36 and is used for crankcases, oil pans, steering-wheel spiders, differential carriers, transmission cases, camshaft housings, hub caps and similar parts. A shrinkage of 0.156 (5/32) in. per foot should be allowed in pattern designs.

SPECIFICATION Nos. 30, 31, 32, 34

Omit aluminum minimum and the percentage designated in the composition limits for each of these alloys. Add as the last item of each of the alloy compositions:

Aluminum Remainder

With the exception of this change, retain the specifications as printed in the present edition of the S.A.E. HANDBOOK.

SPECIFICATION No. 35

Change composition to the following:

Silicon	4.5—6.0
Copper	not over 0.6
Iron	not over 1.0
Zinc	not over 0.2
Manganese	not over 0.3
Magnesium	not over trace
Other impurities	not over 0.3
Aluminum	Remainder

SPECIFICATION No. 78

Change the title of the table of physical properties at the top of p. 505 to read: Physical Properties of Aluminum Flat Sheet.

Add the following paragraph immediately following this table:

Aluminum strip sheet shows the same physical properties as flat sheet, with the exception that in tempers No. 4 and No. 6 the elongations may be slightly lower than those shown for flat sheet.

Replace the table of thickness tolerances with the same tables as given in this memorandum for aluminum-manganese alloy sheet and strip.

Protective Coatings for Aluminum

(*Proposed General Information*)

Following out its policy of providing general information on various non-ferrous materials and plating methods, the Division has developed some data of general interest to the industry on protective coatings for aluminum which is submitted herewith with the recommendation that it be approved by the Standards Committee as General Information for publication in the HANDBOOK.

ALCLAD ALLOYS

Aluminum is highly resistant to atmospheric corrosion and to a variety of chemicals. In general, the resistance is greater, the higher the purity of the metal. The improvement in mechanical properties which results from the alloying of the metal is usually attended by some loss in corrosion resistance. This effect varies with the constituent which is added, and also in some cases with the heat-treatment which the alloy receives.

Aluminum of commercial purity contains a minimum of 99 per cent of that metal, the impurities consisting principally of iron and silicon. This metal is somewhat less resistant to severe corrosive conditions than is high-purity aluminum.

For ordinary atmospheric conditions, commercial aluminum and practically all of its commercial alloys show very satisfactory resistance to corrosion. The alloy containing 1¼ per cent of manganese (Specification No. ——) is at least the equal of commercial aluminum in this respect.

Of the casting alloys, those containing silicon as the hardening element (Specifications Nos. 35 and 37) are the most resistant to salt water. The heat-treated copper alloy (Specification No. 38) follows these very closely and is distinctly superior to the other alloys containing copper (Specifications Nos. 30, 32 and 33). The alloy containing considerable percentage of zinc (Specification No. 31) is the most susceptible to the action of salt water of any of the casting alloys in common use.

The strong wrought alloys (Specification Nos. —— and ——) are not so resistant to highly corrosive conditions as are pure aluminum and the wrought alloy containing manganese.

There are also appreciable differences in this property among the different alloys and in the various tempers. In every case the alloy is in its most resistant state when it has been subjected to the solution heat-treatment alone and has not been subsequently heated to bring about the precipitation heat-treatment or artificial aging.

The most resistant of all of these materials is the alloy containing magnesium and silicon (Specification No. ——) when in the intermediate temper (*W*). This advantage is lost, however, when it is artificially aged to develop its maximum properties (*T*).

The alloy containing copper and magnesium, and manganese, commonly known as duralumin or 17S, is the only one of these alloys which develops its maximum properties on aging at room temperature after the solution heat-treatment. In line with the statement previously made, it is the most resistant to corrosion of any of the aluminum alloys when in their strongest temper.

The choice among these alloys will depend largely upon the nature of the service they are to perform and upon the design of the part.

For comparatively thin, highly stressed sections, such as wing ribs and tubular struts used on aircraft, duralumin or 17S or Alclad 17S is always to be chosen. For heavier sections such as engine crankcases, where superficial surface action would be of little consequence even if it should occur, the other alloys (Specifications Nos. —— and ——) are regularly used with entire satisfaction. Similarly, plates and structural shapes used in motorcoach and motortruck body construction may be made from any of the three alloys.

Even though these alloys are quite resistant to ordi-

nary atmospheric conditions, it is recommended that a protective coating be applied as an added precaution. Aluminum paint made by adding aluminum-bronze powder to a suitable grade of long oil varnish is commonly used. Where the part is in contact with water, bituminous paints may be used. Good results have been obtained on seaplane floats using a zinc-chromate paint and also with red-oxide paint, although comparative corrosion tests indicate that the aluminum paint affords better protection against salt-water corrosion.

It should be emphasized that the preparation of the surface of the metal is as important as the selection of the paint to be used. Work is still in progress by the manufacturers of these alloys and by Government bureaus to determine the best methods of protection.

Up to the present, no method has been devised which gives better results than the anodic oxidation of the surface prior to the application of the paint. This process consists of making the alloy anode in a solution of chromic acid under proper conditions of voltage, temperature and time.

For other less exacting conditions of service than seaplane work, other methods of securing paint adherence may be adequate. The work now in progress may develop methods which are equally effective and at the same time more easily applied.

Taking advantage of the superior corrosion resistance of aluminum of high purity, a process has been developed for manufacturing sheets having a core of 17S (Specification No. ——) with surface layers of pure aluminum alloy and integral with it. This product is marketed under the trade mark Alclad 17S.

Tensile-test specimens of this material exposed for more than 18 months to a standard salt-spray test have shown no loss in tensile strength or elongation. The pure metal not only protects the alloy where it covers it but also protects sheared edges or scratches through the surface layers by its electrolytic action. It has also been found that ordinary duralumin or 17S rivets receive a very considerable amount of protection from contact with the pure-aluminum surface when used with Alclad sheet.

When this material is used for seaplane construction, it is advised that the sheet be given the same measure of surface protection that is given to the ordinary alloy. Service tests may show this not to be essential, but it is considered desirable added insurance of satisfactory performance under such adverse conditions. For the less exacting service of landplanes, Alclad sheet is in use without the protection of a paint coating and is showing entirely satisfactory results.

Screw-Threads Division

PERSONNEL

E. H. Ehrman, *Chairman*	Standard Screw Co.
Earle Buckingham, *Vice-Chairman*	Massachusetts Institute of Technology
A. Boor	Willys-Overland Co.
E. J. Bryant	Greenfield Tap & Die Corp.
Ellwood Burdsall	Russell, Burdsall & Ward Bolt & Nut Co.
Luther D. Burlingame	Brown & Sharpe Mfg. Co.
George S. Case	Lamson & Sessions Co.
R. M. Heames	Victor Peninsular Division, Allied Products Corp.
W. J. Outcalt	General Motors Corp.
L. L. Roberts	Packard Motor Car Co.
S. B. Terry	Pratt & Whitney Co. Division, Niles-Bement-Pond Co.
O. B. Zimmerman	International Harvester Co.

Round Unslotted Head Bolts—Automobile Type

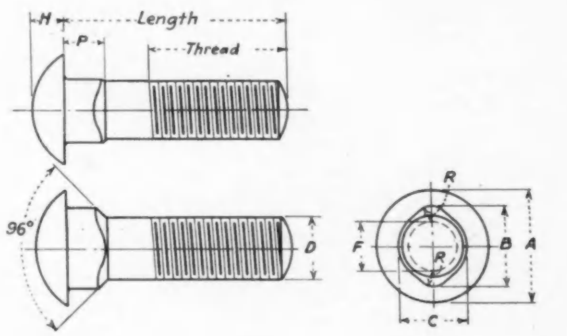
(*Proposed S.A.E. Standard*)

Following adoption by the Society of the standard for Round Unslotted Head Bolts in February, 1929, it was suggested to the Screw-Threads Division that a standard be adopted for the round unslotted-head bolt having an oval or diamond-section neck under the head. It was indicated that this type of bolt is especially applicable in automobile work, particularly where the bolt goes through wood. Preliminary data on a series of sizes of this type of bolt were discussed at the meeting of the Screw-Threads Division in Detroit last January and referred to the following Subdivision that was organized to draft a report!

W. M. Horton, *Chairman*	Lamson & Sessions Co.
A. Boor	Willys-Overland Co.
W. J. Outcalt	General Motors Corp.
O. B. Zimmerman	International Harvester Co.

The accompanying specification was later referred to the Screw-Threads Division for approval and the Division now submits the report for approval and adoption by the Society as S.A.E. Standard. All of the dimensions given other than those relating to the neck under the head correspond to the present standard for Round Unslotted Head Bolts.

At the time of printing this report, the letter-ballot of approval by the Division had not been completed and the report was accordingly printed subject to final approval by the completed ballot of the Division.



DIAMOND-NECK AUTOMOBILE BOLTS

Nominal Size	D		Threads per Inch	A		H		P		C		B		R	
	Major Diameter of Thread			Diameter of Head		Height of Head		Depth of Shoulder		Thickness of Shoulder		Width of Shoulder		Radius of Shoulder	
	Maximum Basic	Tolerance −		Basic	Tolerance ±	Basic	Tolerance ±	Minimum Basic	Tolerance +	Minimum Basic	Tolerance +	Minimum Basic	Tolerance +	Minimum Basic	Tolerance +
No. 10	0.190	0.009	24	0.438	0.010	0.094	0.010	0.188	0.031	0.210	0.005	0.238	0.010	0.047	0.010
¼	0.250	0.010	20	0.563	0.010	0.125	0.010	0.219	0.031	0.270	0.005	0.328	0.010	0.063	0.010
5/16	0.313	0.013	18	0.688	0.010	0.156	0.010	0.250	0.031	0.333	0.005	0.391	0.010	0.078	0.010
⅜	0.375	0.015	16	0.813	0.010	0.188	0.010	0.281	0.031	0.395	0.005	0.468	0.010	0.094	0.010
7/16	0.438	0.015	14	0.938	0.010	0.219	0.010	0.313	0.031	0.458	0.005	0.546	0.010	0.109	0.010
½	0.500	0.015	13	1.063	0.010	0.250	0.010	0.344	0.031	0.520	0.005	0.625	0.010	0.125	0.010
9/16	0.563	0.016	12	1.188	0.015	0.281	0.015	0.375	0.031	0.583	0.006	0.703	0.015	0.141	0.015
⅝	0.625	0.017	11	1.313	0.015	0.313	0.015	0.406	0.031	0.645	0.006	0.781	0.015	0.156	0.015
¾	0.750	0.020	10	1.563	0.015	0.375	0.015	0.469	0.031	0.770	0.006	0.938	0.015	0.188	0.015

Threads are American (National) Standard (Class 2) free fit coarse series.

$A = 2D + 1/16$ in.
$B = 5D/4$

$C = D + 0.020$ in.
$F = 3D/4$
$H = D/2$

$P = D/2 + 3/32$ in.
$R = D/4$

Radius of fillet between body and head shall be 1/32 in. on sizes No. 10 to ½ in. inclusive and 1/16 in. on sizes 1/16 to ¾ in. inclusive.

Threads may be either cut or rolled. When threads are rolled the shank diameter will necessarily be smaller than for corresponding cut threads.